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Safety Last

General Electric Oil Engines for United States Government

By Alfred D. Blake

SYNOPSIS—A high-compression engine of the opposed-piston type running at 500 r.p.m., weighing only 105 lb. per b.hp. and embodying several novel features.

The General Electric Co. has recently completed at its Erie Works seven 150-kw. vertical high-speed high-compression oil-engine generating sets for the United States Government. The requirements of the service demanded lightness and compactness, to which end the two-stroke-cycle opposed-piston design and a speed of 500 r.p.m. were selected, with the generator bolted directly to the engine frame, its bearing being bored concentric with the engine bearings and the flywheel serving as a coupling between the engine and generator; see Figs. 1 and 4.

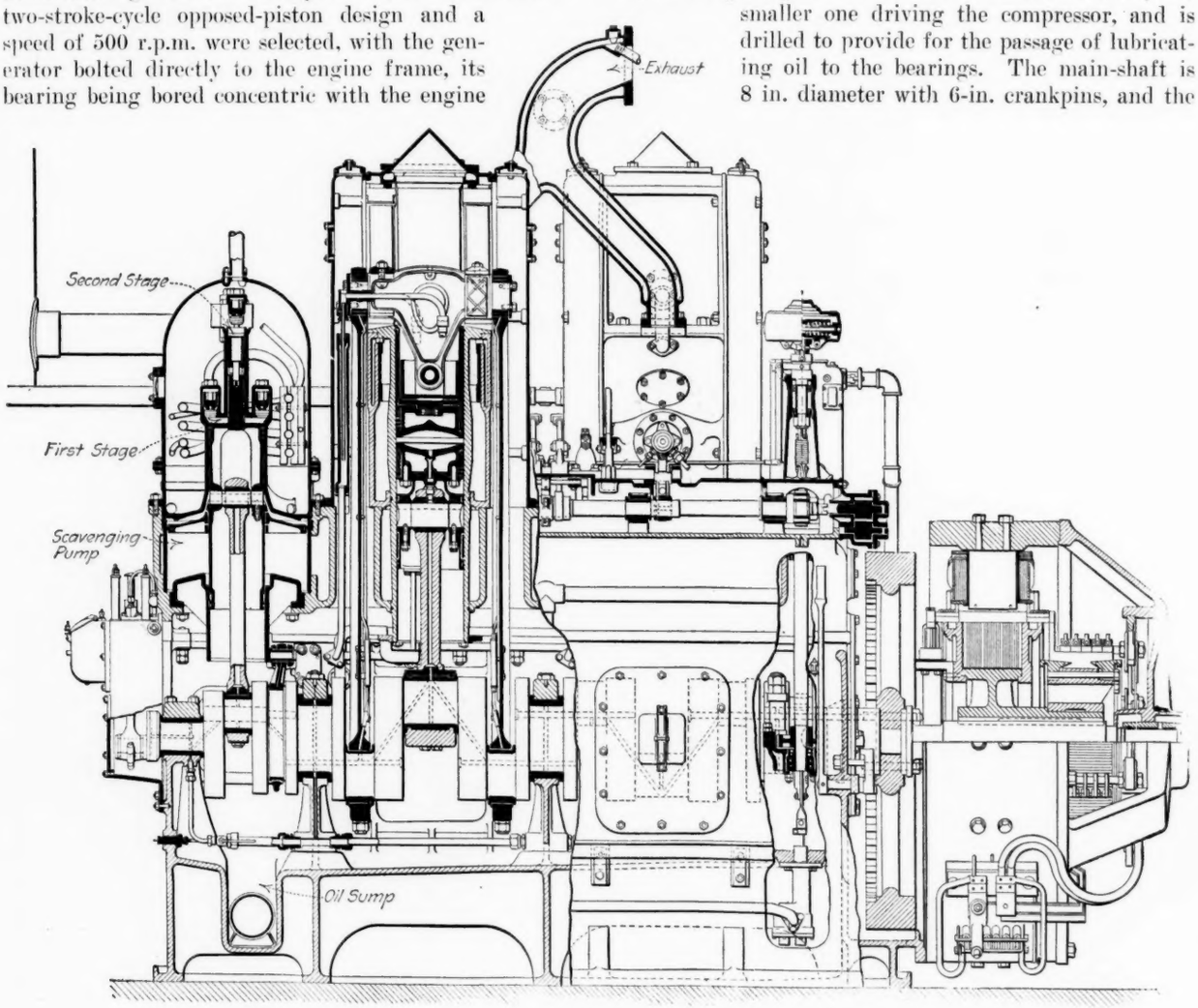


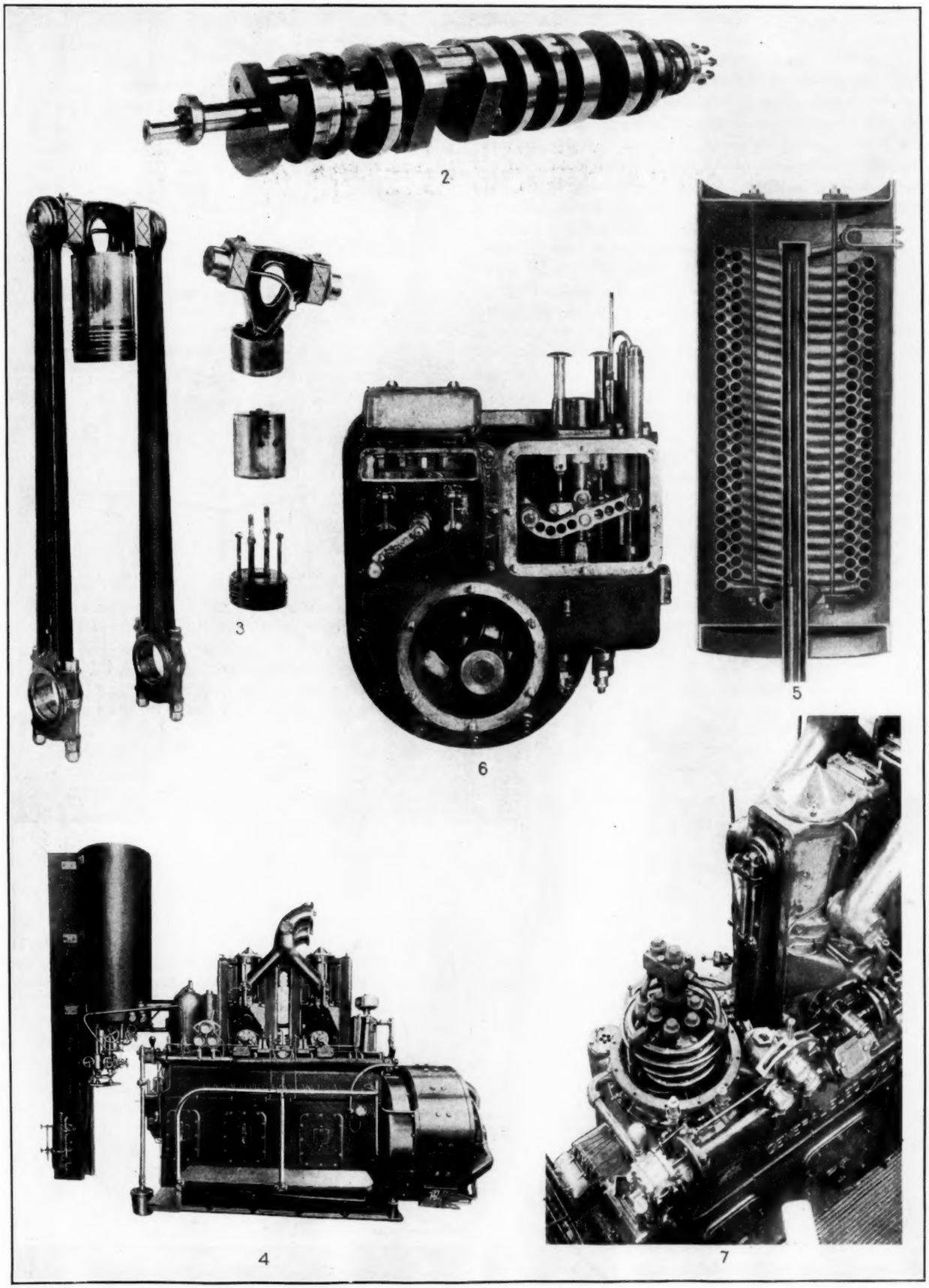
FIG. 1. PARTIAL LONGITUDINAL SECTION, SHOWING WORKING CYLINDER AND COMPRESSOR

These engines are the first of their kind to be built by this company and are the product of nearly three years' investigation and experiment, as a result of which they embody several novel features intended to overcome well-known inherent short-comings of the high-speed two-stroke-cycle Diesel.

An inclosed frame is used with the scavenging-air receiver set on the crank case and carrying the two working cylinders and the air compressor, which is driven from the mainshaft. There are three main bearings liberally proportioned to allow for low bearing pressures and a fourth, smaller, bearing for the compressor; the generator bearing, as previously indicated, is carried on a bracket. Access is had to the bearings by removing the plates on the side of the crank case.

A forged-steel crankshaft having a tensile strength of 90,000 lb. per sq.in. is used. It is in two sections, the smaller one driving the compressor, and is drilled to provide for the passage of lubricating oil to the bearings. The main-shaft is 8 in. diameter with 6-in. crankpins, and the groups of cranks for the two working cylinders are set at 180 deg., those for the upper and lower pistons of each cylinder being displaced 165 deg. The camshaft is driven through spiral gears from a vertical shaft and operates the fuel injectors and starting valves as well as some of the pumps.

The cylinders are of special hard-grained cast iron without liners, the lower ends extending well into the



FIGS. 2 TO 7. DETAILS AND ASSEMBLED VIEW OF GENERAL ELECTRIC HIGH-COMPRESSION OIL ENGINE

Fig. 2—Main crankshaft. Fig. 3—Upper piston and side rods. Fig. 4—Engine complete. Fig. 5—Air-starting coils and blast container. Fig. 6—Fuel pump and force feed lubricator. Fig. 7—Air compressor with dome removed

scavenging-air receiver and thus materially reducing the height of the engine. The exhaust-port bridges are water-cooled.

As will be seen by reference to Fig. 3, the upper pistons are in three parts—an oil-cooled head, a sleeve and a body—an arrangement intended to take care of the heat conditions incident to the two-stroke cycle. Pivoted in the body of the piston is a yoke carrying the crosshead for the side rods, the object being to compensate for any unequal lengths of the side rods due to wear on their bearings. The lower piston is of the trunk pattern of somewhat different design, as shown in Figs. 1 and 8. Its rod is drilled to permit the passage of lubricating oil to the wristpin. The side rods, through which the thrust on the upper piston is imparted to the shaft, are solid but carry pipes for lubricating the upper bearings and wristpins. Both pistons may be removed through the top of the cylinder, the cover of which is merely a conical plate for keeping out dirt and preventing the escape of any fumes that may get past the upper piston.

The scavenging pump is in tandem with the air compressor, as may be seen in Fig. 1. It is double-acting and takes its suction partly from a muffler around the coils in the starting bottle and partly from the crank case so as to draw in any gases that may escape past the lower pistons. It has an eccentric-operated balanced piston valve controlling the suction and the delivery on either side to the scavenging-air receiver at about 6 lb. A relief valve set at 10 lb. is attached to the receiver to take care of any excess pressure. The scavenging ports in the cylinder are set so as to impart a spiral motion

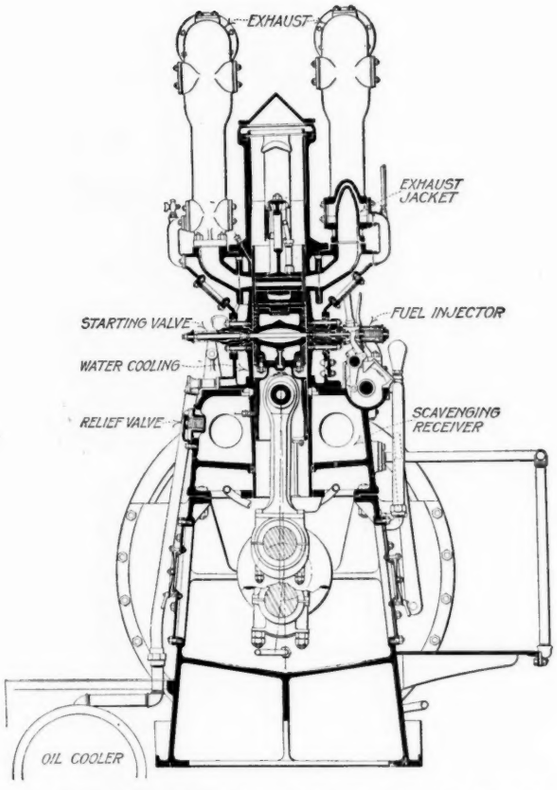


FIG. 8. TRANSVERSE SECTION THROUGH ENGINE, SHOWING THE GENERAL CONSTRUCTION

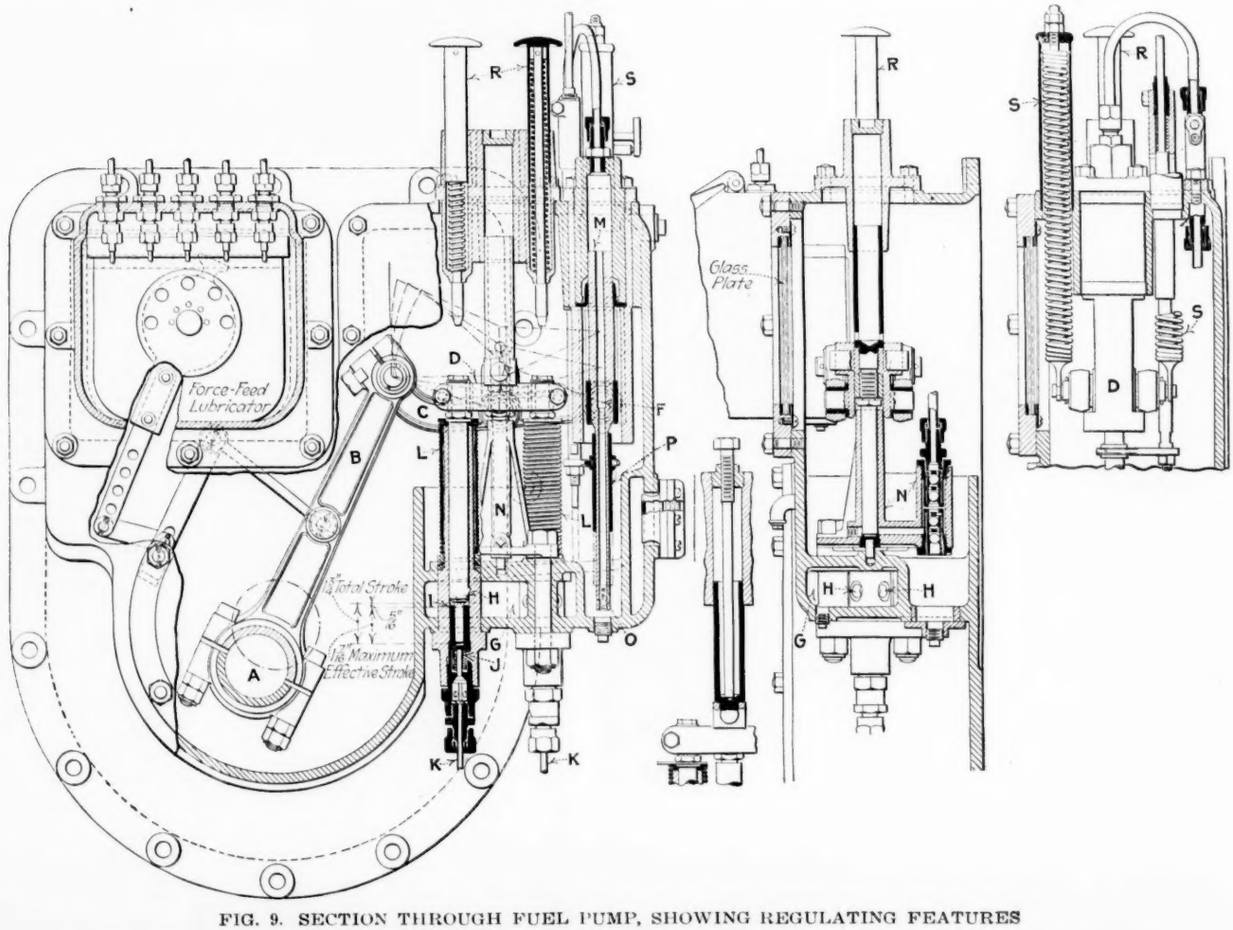


FIG. 9. SECTION THROUGH FUEL PUMP, SHOWING REGULATING FEATURES

to the air and are uncovered 15 deg. later than the exhaust ports, which permits a reduction in terminal pressure sufficient to allow an inrush of scavenging air.

The air compressor is of the two-stage single-acting type, with intercooler and aftercooler. These consist of copper coils located in the dome and entirely surrounded by water. Each is provided with relief valves. A clear idea of this arrangement will be had from Fig. 7, which shows the top of the compressor with the dome removed.

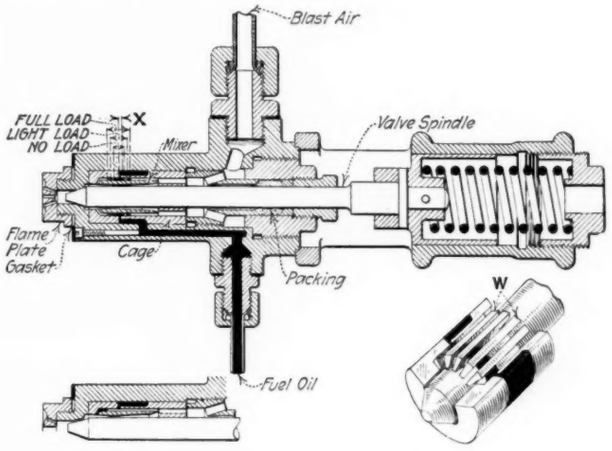


FIG. 10. SECTION THROUGH FUEL INJECTOR

As an additional precaution a safety diaphragm is provided in the dome. There are four suction and two discharge valves on the low-pressure stage, and one suction and one discharge valve on the high-pressure stage. These are all of the disk type without springs, are of the same size and interchangeable, it being necessary only to reverse the valve to use it on either the discharge or the suction.

Air is compressed in the first stage to about 100 lb. and in the second to 750 to 1,000 lb., at which it is delivered into the air-blast and starting container. The compressor suction is normally taken from the silencer, but at the option of the operator may be taken from the scavenger receiver, in which case it passes through an oil separator.

Instead of the usual form of bottle for the storage of the injection air and starting air, the latter consists of a coil of 2-in. tubing surrounding an upright member also made of steel tubing. This arrangement provides an exceptionally safe way of storing the high-pressure air. As an additional measure of safety the whole is surrounded by a steel cylinder with a perforated head, which would protect persons about the engine should a coil perchance let go. The coils have relief valves and control valves at the bottom, and the whole is mounted very compactly near the engine; see Fig. 4.

The starting coils have a capacity of four starts. As the service conditions that these particular engines will have to meet do not call for frequent starting and stopping, the starting system has been simplified. The engine is barred over to the starting position, the air is turned on the system, and by throwing the control lever to the starting position, the proper cam is brought in contact with the air starting valve. Filling of the starting coils is effected by controlling the suction of the air compressor.

There are two fuel pumps, one for each cylinder, carried on the crank case and driven by a crank on the end of the mainshaft; see Figs. 6 and 9. They operate on the constant-stroke variable-zone principle and contain no suction valves and no packing, the latter being a feature that will appeal to operators.

Taking up the fuel pumps in detail, reference will be made to Fig. 9. Here A is the crankpin, which actuates the pump plungers through the levers B and C and the yoke D. The oil flows by gravity to the receiver G and through the holes H surrounds the plunger end, which is of tool-steel ground to a glass-hard finish. The fulcrum F of the lever C is raised and lowered through the action of the governor, and although the stroke is practically constant, the zone of its action varies with the position of the fulcrum F. This is what is meant by the "constant-stroke variable-zone principle." In this way measured quantities of fuel to suit the load are "punched out," so to speak, by the plunger at I. Two discharge valves in series are provided at J, and the pipes KK lead to the two fuel injectors. The pump cannot become airbound through low oil, as any air will rise to the top of the receiver. A removable glass cover on the front of the pump permits its operation to be under the eye of the attendant.

The resistance encountered in forcing out the oil produces a reaction tending to force the fulcrum F upward; this is aided by the springs LL, whose other function is the return of the pump plungers. To increase the quantity of oil to the fuel injectors, the fulcrum must be forced down by the plunger, which is under oil pressure from a "regulating pump" N (see transverse section, Fig. 9) between the fuel pumps and operated by the yoke D. This regulating pump has both suction and discharge valves, and the oil delivered by it tends to force down the plunger M and consequently the fulcrum F until permitted to escape through a small port O in an extension of M, when uncovered by the edge of the "chasing sleeve" P, which is connected to the governor. When the port is thus uncovered, the pressure is relieved and the reaction of the fuel pumps, aided by the springs,

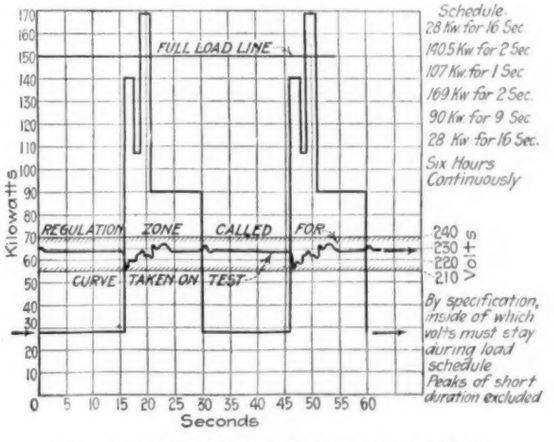


FIG. 11. CHARACTERISTIC LOAD CURVE

causes the fulcrum F to rise until the port O is again covered. A balance is effected in this way depending upon the position of the governor, which is of the centrifugal type, whose internal forces are self-contained. This arangement relieves the governor of all strain. At no-load or very light loads, when the reaction produced by the pump plungers is light, the springs SS come into action to raise the fulcrum. The governor is driven through a spiral spring to eliminate hunting.

By opening a bypass valve in the regulating-pump delivery line, the pressure on the plunger *M* is released and the supply of fuel is immediately cut off, due to the fuel-pump plungers operating in a noneffective zone. Another bypass valve is connected with the emergency stop on the governor, set for 600 to 625 r.p.m., or it may be worked by hand; in either case the engine is brought to a stop. The plungers *RR* are for hand priming.

Special Design of Fuel Injectors Used

A horizontal closed-type fuel injector of special design is used. This is shown in section in Fig. 10. A measured quantity of oil is delivered by the fuel pump and is forced through the small clearance *X* (about 0.007 in.) forming slugs of oil in the channels *W* of the mixer. Before the spindle is opened, these slugs are subjected to a balanced pressure on both sides, but as soon as it is opened, a venturi action is set up and the slugs are picked up and atomized by the blast. The radial holes in the flame plate are merely for directing the spray.

It is claimed that, owing to this form of injector, the blast pressure adjusts itself to the load. That is, if the suction valves on the compressor are set for light load, heavier variable loads will be taken care of and if set for full load, lighter variable loads will be taken care of. This is because the resistance offered the air through this injector is due almost entirely to the oil to be moved; if it be a large quantity, the pressure builds up behind it, and if the quantity be small, the rapid escape of air will reduce the pressure automatically.

The injection valve is normally timed to open 5 deg. ahead of the upper piston center and 20 deg. ahead of the lower piston center, or 12½ deg. ahead of the point of maximum compression. The total period of opening of the fuel valve is 63 deg.

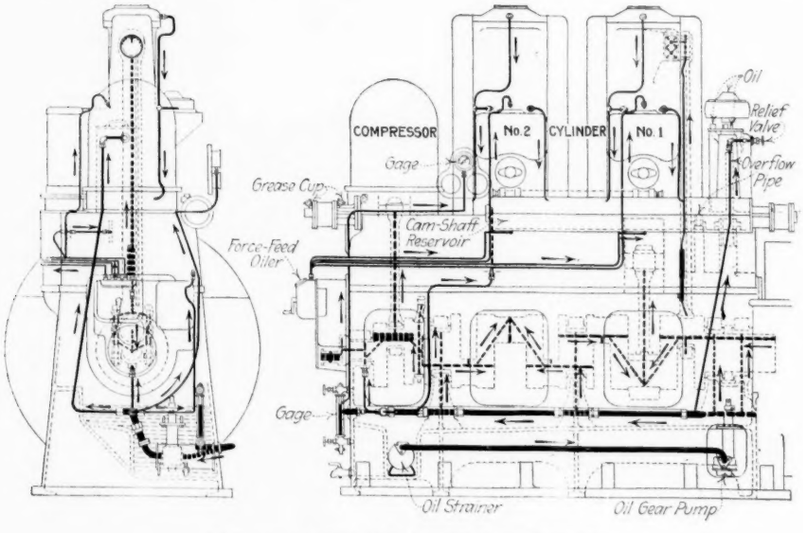


FIG. 12. DIAGRAM OF OILING SYSTEM

In service these engines will be called upon to carry an extremely variable load the character of which is indicated in Fig. 11. The voltage regulation specified and that actually obtained by test under these load conditions are also given.

From a strainer in the pump beneath the crank case the oil flows to a gear pump that delivers to the main bearings, thence through the cranks to the crankpin bearings, and up the hollow connecting-rods to the wristpins of the lower pistons. In the upper wristpins the oil is forced from the crankpin bearings up a pipe carrying a check valve on the side rods to the crosshead and thence to the upper wristpins; see Fig. 12. In all bearings the oil is delivered to the side under pressure.

A branch from the pipe supplying the main bearings goes to a priming tank for the piston-cooling system, at the generator end of the engine, and an overflow from

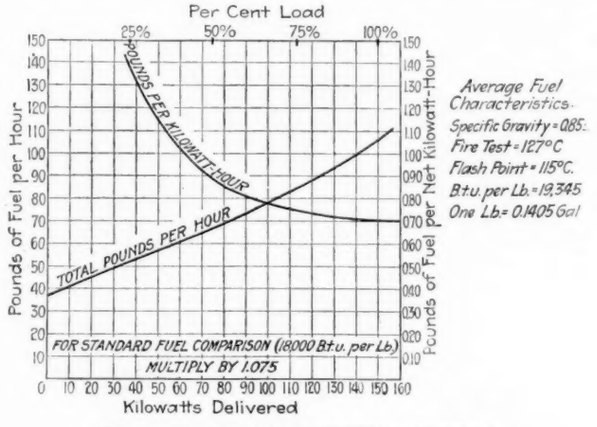


FIG. 13. FUEL-CONSUMPTION CURVES

this tank leads to the pump. The camshaft runs in oil, the level being maintained by an overflow to the pump. The cylinders are lubricated by a force-feed lubricator mounted beside the fuel pumps (Figs. 6 and 9).

Oil and Water Cooling Systems

The pistons are oil-cooled under a pressure of 60 to 70 lb. From the priming tank, previously mentioned, the oil flows to a centrifugal circulating pump driven from the end of the camshaft, to a main from which branches lead to telescopic connections, thence to the pistons. The returns are brought to a collecting pipe at the back of the engine, thence to an oil cooler consisting of a number of spiral copper coils set in a cylinder through which water is passed. From the cooler the oil returns to the priming tank, and all makeup is furnished from the lubricating system, as previously explained.

The path of the cooling water is from the strainer through a check valve to the circulating pump, which is of the rotary plunger type. From the pump it is forced to the compressor dome, then to the oil cooler, through sight-flow indicators to the cylinder exhaust belts, to the exhaust manifold, and is finally sprayed into the exhaust pipe.

The engine is exceptionally light for the output, weighing 21,221 lb. exclusive of flywheel and generator, 22,896 lb. including the flywheel, and 32,105 lb. including both flywheel and generator. This is equivalent to about 105 lb. per normal brake horsepower for the engine and flywheel and 214 lb. per kilowatt for the complete unit.

Fuel-consumption figures as obtained on the official tests are given in Fig. 13.

Coal More Economical Than Oil

By James Ross

SYNOPSIS—A plant having three 500-hp. water-tube boilers using oil fuel costing 87c. per bbl. changed to stoker fired coal purchased for $2.75 per ton and saved nearly $5,000 in nine months.

A certain plant in New Orleans consists of three water-tube boilers of 500 hp. each. Two years ago this plant was burning oil under its boilers at a contract price of 87c. per bbl., the amount of oil consumed being about 25,000 bbl. per yr. When the contract expired, oil had risen to $1.16 per bbl., and as this price was considered prohibitive, it was decided to investigate the use of coal.

The heating value of the oil used was 18,500 B.t.u. per lb., and each barrel weighed 320 lb. The number of heat units in a barrel of oil therefore averaged 5,900,000. If coal were substituted containing 12,500 B.t.u. per lb., or 25,000,000 B.t.u. per ton, one ton would be equivalent to 4.23 bbl. of oil. Owing to the losses incident to the handling of coal, it was estimated that the heating value of one ton as purchased would be about equal to that of four barrels of oil; and therefore, the amount of coal that would have to be burned to replace the oil would be 6,250 tons per year. At $2.75 per ton delivered, the fuel cost would then be as follows:

Burning coal at $2.75 per ton	$17,200 per yr.
Burning oil at 87c. per bbl.	21,750 per yr.
Burning oil at $1.16 per bbl.	29,000 per yr.

In this comparison of costs it was assumed that coal would be burned as efficiently as oil, as it was believed that the installation of a good type of stoker would make such results attainable. It was also believed that, if coal-handling apparatus were installed, the cost of firing coal would be no greater than that of firing oil; and since the price of furnace maintenance with the two fuels should not differ greatly, the difference between the estimated fuel costs would be a net saving in favor of coal. This saving amounted to $11,800 per year when compared with oil at $1.16 per bbl.; and it was thought that such a saving would fully justify the expenditure of $20,000, which was the estimated cost of a stoker and coal-handling system.

A Stoker is Installed

After a careful investigation of the various stokers on the market, it was decided to install one of the well-known underfeed types under two of the boilers and arrange the third boiler for hand firing. These stokers were guaranteed to show a combined boiler and furnace efficiency of 72.5 per cent. at normal rating and of 70 per cent. at 150 per cent. of rating, and were further guaranteed to be capable of developing 850 hp. from each of the 500-hp. boilers.

Among the reasons that led to the installation of these stokers, in addition to the guarantees, were the following:

1. Air for burning the coal is supplied by a forced-draft fan, which is directly connected to the same engine that drives the stoker. For this reason the amount of air supplied is always directly proportional to the quantity of coal fed, insuring the proper supply of air for combustion under all conditions of load.

2. The stoker is self-cleaning; the coal as it is burned is worked gradually to the back until it reaches the dump plate, from which the ash is dropped into the pit below. This obviates the necessity of opening fire-doors for cleaning, which always entails a loss in efficiency and a drop in capacity while the fires are being cleaned, leveled or raked.

3. The guaranteed operating and maintenance cost of the stoker was low. The amount of steam required to run the driving engine at normal load was not to exceed 3 per cent. of the total steam generated, and the upkeep cost of each furnace was not to exceed $75 per year.

These advantages were considered important enough to justify the purchase of the type selected in spite of the fact that the price was considerably higher than that of others and that the cost of installing them would also be higher. Each stoker has an active grate area 9 ft. wide by approximately 9 ft. deep.

Along with the stokers an automatic coal- and ash-handling system was installed. The coal, after passing through a crusher, is carried to an overhead bin, from which chutes conduct it to the stoker hoppers. The same conveyor carries the ashes from the ashpit to an overhead bin, from which they can be loaded into carts.

This plant has been operating under new conditions for two years, and the results have fully justified the action in changing over from oil. A comparison of operating costs before and after making the change is of interest:

The labor cost with the stokers is the same as it was with oil, two firemen being employed at $60 per month each.

Maintenance Cost Less with Stokers

The maintenance cost with the stokers is less than it was with oil. The stokers have averaged $50 per year to keep in repair, and there has been no expense so far in connection with the brickwork. With the old oil burners the expense of upkeep including brickwork was $197.25 per year.

It is in fuel cost, however, that the greatest saving has been made. On first starting up the stokers, tests were run on various grades of coal, varying in price from $3.25 per ton down, to determine the grade best adapted to the conditions. The results of these tests led to the adoption of a coal worth $2.75 per ton, as this was found to give the best results with the stokers. In the nine months from Jan. 1 to Sept. 30, 1913, the cost of coal was $11,539.82. In the corresponding nine months of the preceding year the cost of oil was $16,468.69. This shows a saving of $4,928.87 in nine months, or of about $6,500 per year, as compared with oil at the old contract price of 87c. per barrel. The reason that a period of only nine months is here compared is that it was for only this length of time after the stokers were installed that the load was the same as in the preceding year when burning oil. In October, 1913, an additional load was placed on the boilers, which prevents a direct comparison of fuel costs over a longer period, but even with this increase in load, the fuel cost is much less than formerly.

Symposium on Welding as Applied to Boilers

SYNOPSIS—Some prominent men in the boiler field express themselves on electric and oxyacetylene welding as applied to high-pressure boilers.

[As the financial and physical success of welding, as applied to various kinds of work and metals and to repairs in plants, increases, all interested in steam boilers and other power-plant pressure vessels more and more speak and wonder about welding as applied to such vessels, particularly with reference to safety. One hears it said that an effort will be made before this or that legislature or board to permit it, while before some other action forbidding it is assured. It is whispered that folks from abroad who seek to apply welding, particularly to boilers, can demonstrate the safety of their work and show worthy data to further establish their claim.

In view of the evident confusion of claims the opinions of responsible and experienced men relative to the safety of such work are desired by all makers and users of boilers and, of course, by the public.

The opinions given here are few. We know there are many other representative men whose views would be appreciated. *Power* would be glad to have them submitted.—Editor.]

William H. Boehm, superintendent, steam-boiler and flywheel departments, Fidelity and Casualty Co., New York: It has been our practice to advise against the use of oxyacetylene welding methods for the repairing of steam boilers. On this account we have no accumulated experience with such welds upon which to form a general opinion as to the safety of the process. Our attitude is merely this: That the character of the weld depends so largely upon the skill of the workmen that any assumed strength is a decided uncertainty.

We do not care much how weak a structure may be, because then we can determine that load on the weakened structure which will allow the proper factor of safety. That is to say, we are not at all afraid of known weaknesses, but we are afraid, and naturally should be afraid, of an uncertain strength, and it is this uncertainty that prejudices us against the use of the method.

Welds in Tension Not Insured

S. F. Jeter, chief engineer of the Hartford Steam Boiler Inspection and Insurance Co.: We have little actual knowledge of the making of the different kinds of welds, but we have had considerable experience in observing their action in service. Pressure welds are accepted by us for insurance, and since we have accepted the Boiler Code of the American Society of Mechanical Engineers as our standard for new work, their conclusions as regards the strength of such welds, as expressed in paragraph 186, page 45 of the Code, represents our position in the matter. We do not accept autogenous welds of any character where they are subjected to tension and where the strength of the weld directly affects the safety of the boiler.

We have had considerable experience with such welds on the shells of boilers and have found them to be unreliable. Of course, as everyone knows who has had any connection with autogenous welding by the flame method, the skill of the operator in making the weld is an important factor, such skill determining whether the proper mixture of gas is obtained and also whether oxides of the metals are left in the weld to affect its strength; but aside from the most skillful operation we have found that where attempts are made to weld any considerable length in, say, a shell of a boiler, by the autogenous method, internal stresses are set up by the act of welding that cannot be estimated, and while skillful manipulation can no doubt greatly reduce these stresses, we feel that they cannot be eliminated unless the surrounding parts are extremely flexible, and this matter of flexibility, of course, cannot be arranged for in repairs.

Believes Welding Coming Practice

Frank L. Fairbanks, chief engineer, Quincy Market Cold Storage and Warehouse Co., Boston, Mass.: In relation to the matter of acetylene welding, it may interest you to know that I have just returned from the manufacturers after testing two acetylene-welded 200-ton (refrigeration) shells which had been tested by the manufacturer twice and had been pronounced perfection itself. I insisted upon seeing the test applied, however, and while the shell stood the pressure test all right, when a 14-lb. hammer was applied to one of the girth seams under pressure, this seam let go for something like 18 in., and it is likely that had it been followed up, the break would have continued completely around the shell. The only reason that the entire shell did not let go is the fact that some 200 tubes rolled into the heads were really carrying the stress of the sheet; the welding was little more than making tight.

Acetylene-welded and electric-welded vessels will some day be possible with such a high factor of safety that it will be the only way to make up shell vessels. This will not come until each manufacturer stops condemning the other fellow's work, claiming that his own is the only good weld, and it will not be until the whole device to be welded is heated beyond the critical temperature and the welding done under this condition.

Present Welds Are Uncertainties

Frankly, in talking with disinterested men who have become competent by experience to judge, I have yet to find such a man to contend that acetylene welds as at present made are anything but uncertainties, especially in large vessels.

Power can do no better service to those interested than to induce men who are competent to thresh out this question.

J. C. McCabe, chief inspector, City of Detroit, Mich.: In response to your request for an expression from me as to the desirability of autogenous welding about the steam boiler, I submit the following:

The engineer and mechanic are naturally restive when called on to perform operations in production or repair of an article in commerce, when it is apparent that the methods used are laborious, indirect and wasteful as to

time and material. The time when the sheet-metal worker was required to fashion a pail, kettle or similar vessel from the sheet and join the various parts with double seams and solder is well within the memory of most of us. Much of this work is now done by the press and in a much more desirable style. The assembling of the sheets of a steam boiler with cover plates and rivets will, in time, be as obsolete as the double seam referred to.

In the consideration of autogenous welding, where the plates joined are subjected to varying temperatures, it is important that the metal fused for joining the plates to be welded conform as nearly as possible to the physical characteristics of the plates in question. If the elastic properties or the coefficient of expansion vary to a large extent, trouble is sure to follow under variable temperatures.

In several instances we have allowed the repair of cracks on heads of internally fired boilers, the cracks in question being the natural result of repeated stresses resulting from temperature differences. The welding in each case failed to hold. This failure should not be taken as evidence of the inutility of autogenous welding. If the plate itself could not stand such service, why expect the weld to do better?

Effect of Fusion Should be Known

It would seem to the writer that the proper course for the advocates of autogenous welding is to get down to brass tacks and show the properties of materials used in welding and the effect of fusion on the same. Coupons should be built up by fusion of welding material under conditions as nearly as possible comparable with the actual work. This office has been called on to approve electric welding in certain low-pressure heating boilers. The original samples submitted were made by the carbon-arc process. The work was so poor and undesirable as to be rejected. Further samples were submitted where the welding was done by the metal-pencil-arc method with satisfactory results for the case in question.

The following tests were made from coupons built up by fusing the Swedish iron rod, or the pencil-electric-arc method:

Dimensions of bar, in.	1.005x0.325
Elastic limit per square inch, lb.	27,709
Tensile strength per square inch, lb.	47,763
Elongation in 2 in., per cent.	10.93
Reduction in area-flaws at break, per cent.	3.98

Second test, same kind of bar:

Dimensions of bar, in.	1.009x0.325
Elastic limit per square inch, lb.	29,703
Tensile strength per square inch, lb.	46,904
Elongation in 2 in., per cent.	6.64
Reduction in area-flaws at break, per cent.	4.97

The following tests were made from bars using the carbon-pencil-arc method:

Dimensions of sample, in.	1.25x0.335
Elastic limit per square inch, lb.	15,333
Tensile strength per square inch, lb.	22,593
Elongation in 2 in., per cent.	0.78
Reduction in area-flaws at break, per cent.	1.79

Dimensions of bar, in.	1.013x0.346
Broke at, lb.	16,980
Flaws at break; slight reduction of area and elongation.	

The following test was made from a built-up coupon acetylene-welded, the wire being 0.15 in. diameter, Swedish iron rod having the following physical characteristics:

Elastic limit per square inch, lb.	28,396
Tensile strength per square inch, lb.	46,705
Elongation in 10 in., per cent.	20

The coupon from the foregoing metal was fused, or built up, in a channel 7/8x5/8 in. milled in a bar of iron. The sides were afterward planed off, leaving the built-up section 3½ in. long, which was welded to pieces of bar to be of convenient length for testing. The test results showed:

Dimensions of bar, in.	0.762x0.5
Elastic limit per square inch, lb.	34,120
Tensile strength per square inch, lb.	48,031
Elongation in 2 in., per cent.	15.6
Reduction in area, per cent.	36.9

In the second acetylene-weld test the wire used was copper-coated steel 0.156 in. diameter, and tested as follows:

Elastic limit per square inch, lb.	57,322
Tensile strength per square inch, lb.	64,918
Elongation in 10 in., per cent.	1.4

The coupon built up from this steel wire was made in the same manner as described for foregoing coupon. The test showed:

Dimensions of bar, in.	0.757x0.500
Elastic limit per square inch, lb.	30,647
Tensile strength, lb.	49,934
Elongation in 2 in., per cent.	32.81
Reduction in area, per cent.	39.34

In my opinion the results of these tests indicate that it is possible to make dependable autogenous welds.

Another acetylene-welded bar built up from a rod gave the following results:

Dimension of bar, in.	0.503x0.376
Bar failed at 24,273 lb. per sq.in. Failure occurred at weld joining built-up portion to extension pieces. Crystals at break. Manipulation of work evidently defective.	

It is possible to make an autogenous weld that will show considerable strength, but what is wanted is a weld wherein the materials, after processing, conform to the physical characteristics of the plate and are reliable. From my present knowledge and experience with autogenous welding, I could not approve it in tension in any place where the failure could do any damage.

Dwight CO_2 Indicator

An unusually simple device for indicating the percentage of CO_2 in the flue gases has just been perfected by the Dwight Manufacturing Co., of Chicago. It consists of a receptacle to hold the charge of gas, with a reservoir in the base for the chemical and a gage on top which is calibrated to give directly the percentage of CO_2 in the gases. It takes but a minute to make a test. To get a fair sample, the gas is drawn through the device during the greater part of this period, and the balance of the time is required to read the gage. As the instrument is made of cast metal, it is fit for boiler-room use. There is nothing mysterious about its operation, and a reading direct from a gage will be appreciated by the average fireman.

To the top of the gage the instrument stands 10 in. high, the base is 5 in. outside diameter, and the upper compartment has one-half this dimension. The chemical is a strong solution of caustic potash, which absorbs almost immediately the CO_2 in a given volume of gases. To prevent action while a sample is being drawn into the upper compartment, the caustic potash is covered by a stratum of light-gravity mineral oil, which is not affected by the chemical.

Enough solution for 300 tests along with the oil is poured into the reservoir in the base through the filling and vent cock shown near the bottom of the instrument. The oil, being lighter than the caustic potash solution, rises to the top to form a protective layer while the sample is being drawn into the upper compartment by the bulb shown in the photograph. When a fair sample has been

collected, the inlet and vent cocks are closed, the instrument is shaken to bring the caustic potash in contact with the gas and absorb the CO_2. The action is quick, so that the percentage of CO_2 can be read immediately on the dial. As soon as the instrument is placed upon its support, the oil again rises to the surface and the indicator is ready for another test. Any thin film of caustic potash that may adhere to the walls of the upper compartment is neutralized at the beginning of the next test by the gas that is pumped through the vessel before the cocks are closed on the sample to be tested.

The operation of the indicator depends upon Dalton's law of gaseous pressures, which states that every part of a mass of gas inclosed in a vessel contributes to the pressure against the sides of the vessel the same amount that it would have exerted by itself had no other gas been present. Consequently, if a certain constituent of the gas, the CO_2 in this case, is absorbed, a corresponding

THE DWIGHT CO_2 INDICATOR

part of the total pressure on the walls of the vessel is removed, and the reduction in pressure indicates the percentage of CO_2 contained. The scale of the indicator is calibrated to read CO_2 percentages directly.

In tests on known percentages of CO_2 and in comparison with the Orsat, the new indicator has proved to be surprisingly accurate. It has the advantage in its favor that the inaccuracy due to the personal element in conducting a test is eliminated.

Care of Thermostatic Traps

By D. N. Crosthwait, Jr.

Thermostatic traps have been in use for a number of years, but it is doubtful if any piece of apparatus so extensively used is more generally abused. With modern methods of manufacture and the knowledge of the properties of materials and chemicals, a thoroughly reliable article has been made possible.

Notwithstanding this, the trap must have the proper conditions to work under, and it should not be abused. Frequently the failure of a heating system to work properly is blamed on the thermostatic trap when it has nothing whatever to do with the trouble. It is not the present purpose to deal with heating-system troubles, but to point out the proper attention to be given the trap.

Thermostatic traps are generally recommended for use at pressures below a certain given maximum, and this maximum is about ten pounds. The pressure given should never be exceeded in operation. One of the best ways of insuring the satisfactory operation of traps is not to tamper with them, as nearly always they are correctly assembled at the factory and are not adjustable. When it becomes necessary to remove the cap and disk for the purpose of cleaning, the disk should not be removed from the cap nor should its position be changed in any way. When exhaust steam is used for heating, it is often found that after three or four seasons' service, the trap will collect quite a bit of oil, but a thorough washing in gasoline or kerosene is all that is necessary to restore the trap to perfect working order.

Do not use hot water for this purpose, because there is a possibility of ruining the trap. Do not allow the traps to be subjected to a higher steam pressure than that for which they are designed, and do not meddle with them unless it is certain that they are the cause of the trouble.

A Change for Better Still

By G. B. Longstreet

A sequel to the verses by R. T. Strohm in March 28 *Power*.

Yes, we've listened to the voice of fickle Fashion,
And her oft and various changes bring us glee,
Such as sending to the junk pile lever safeties
(And it serves 'em right, that's where they ought to be).
There's another change she's surely going to bring us,
For she's harnessed with that war-horse "Safety First,"
And they're tugging at their load with all their power,
To get lap-seam boilers out before they burst.

But their pull is sure to be a steep and stiff one,
For it's hard to make some people understand
That a lot of dangerous weapons of destruction
Are in all the nooks and corners of our land.
True it is, that some of them have hung together,
They have done their duty oft and done it well,
But with every one the chance is always open
To possess a hidden crack beneath its shell.

Now it isn't often that a butt joint fails us,
And as yet it seems to be the safest way
Just to butt the edges each against the other,
Then to strap and rivet them so that they'll stay.
To accept a lap joint now would be an error
That would help to keep us in the same old rut,
So let us remember this when buying boilers
And have each lap seam supplanted by a butt.

Now if every engineer will be a driver,
Taking Safety First and Fashion as his steeds,
With a footing of good able legislation,
They'll supply him all the power that he needs
To uproot and clear away this dire menace—
The old boiler with its seams concealing flaws;
For although we may not do it with persuasion,
We sure can and will do something through our laws.

Valve Details of Univalve Unaflow Steam Engine

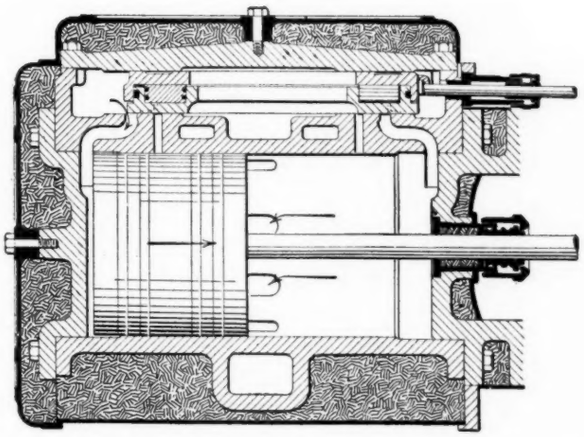


Fig. 1. Head end: Admission on dead center. Frame end: Exhaust through central exhaust ports

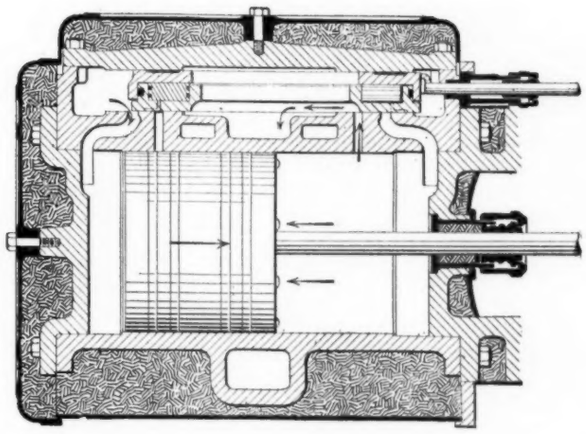


Fig. 2. Head end: Continuation of admission. Frame end: Exhaust through central exhaust ports about to close

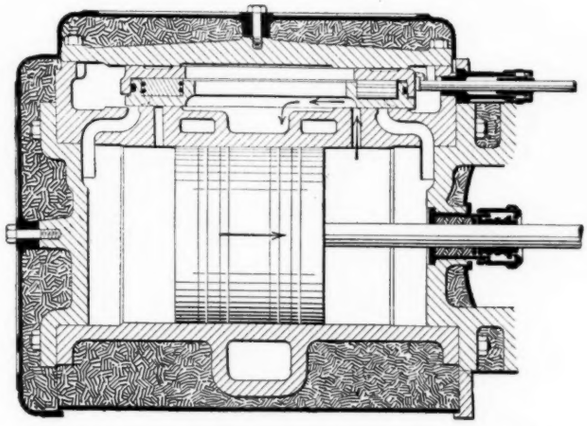


Fig. 3. Head end: Closing of admission port and beginning of expansion. Frame end: Displacement of trapped vapor through open auxiliary exhaust port

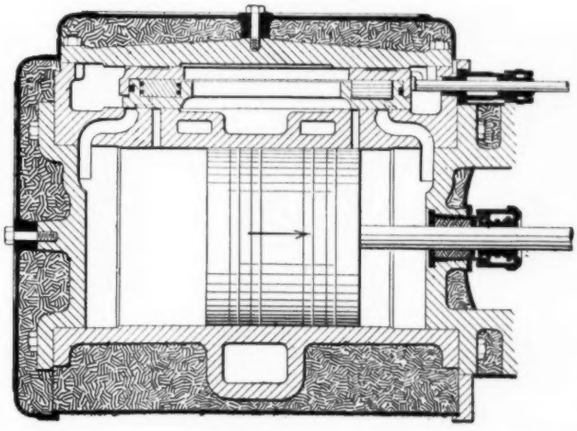


Fig. 4. Head end: Continuation of expansion. Frame end: Commencement of compression by piston covering auxiliary exhaust port

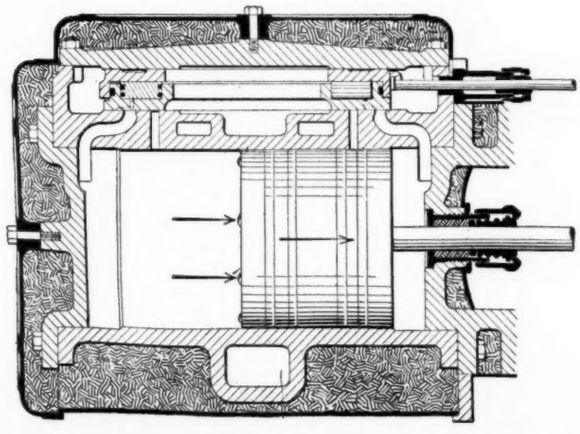


Fig. 5. Head end: Beginning of exhaust through central exhaust ports. Frame end: Continuation of compression

The main characteristic of the univalve unaflow engine is its economy in steam consumption, not only at full load but down to one-fourth load. Another feature is the simplicity of the valve gear, there being only four joints in it.

The unaflow engines as used in American practice generally are operated noncondensing, and it was desirable to delay the beginning of compression without the addition of clearance and without destroying the thermal advantages of the unaflow principle, by placing auxiliary ports at that point in a unaflow cylinder where compression usually begins in a noncondensing counterflow engine. A separate valve for each admission and each auxiliary exhaust port is used. This allows the auxiliary exhaust valves to be thrown in and out of operation and permits the engine to operate both noncondensing and condensing with small clearance.

The cuts, Figs. 1 to 5, illustrate the operation of the valves and piston of this engine, which is manufactured by the Skinner Engine Co., Erie, Penn.

The Jones-Patchel Turbine

Tests have been conducted on the turbine,[1] Fig. 1, invented by E. G. Jones and T. T. Patchel, both of Cape May, N. J. The turbine is a combined reaction and jet type, differing materially from those now on the

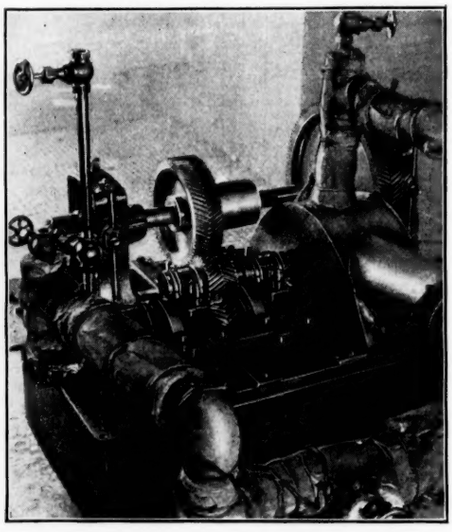


FIG. 1. JONES-PATCHEL TURBINE

market. There are two rotors running in opposite directions, one running inside the other (see Fig. 2). Steam enters the small rotor through the hollow shaft, while steam from the inner rotor reaches the inside of the outer one, which also gets steam on the outside from a pipe coming in through the top of the casing. The course of the steam is shown in Fig. 2, which shows

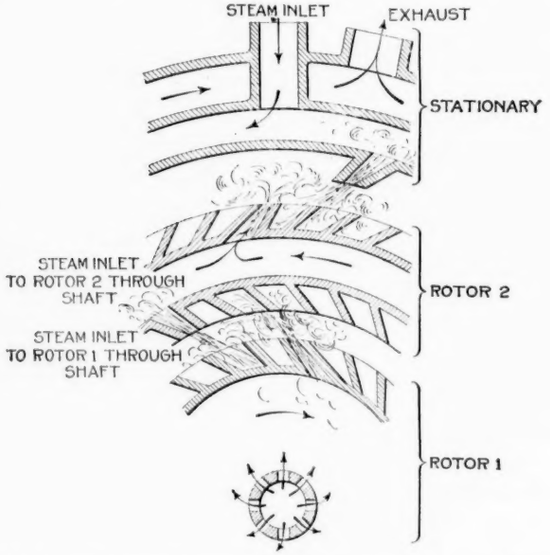


FIG. 2. SECTIONAL ELEVATION OF PART OF TURBINE, SHOWING COURSE OF STEAM

that besides the steam inlet in the shaft there is also one through the top of the casing. The clearance between rotors is 3/8 in.

After the steam is finally expanded, it escapes over the ends of the rotors into the very outer or exhaust casing.

[1]Patent No. 1,079,177.

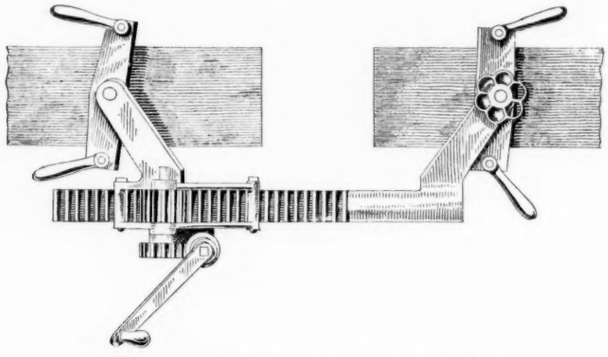
Some of the performance data furnished by the inventors follow: Steam pressure, 82.5 lb.; brake horsepower developed, 28.06 at 4,548 r.p.m.; steam per brake horsepower, 63.48 lb. per hr.

Clefabco Belt Adjuster

Most engineers are familiar with the old type of belt clamps and the time required to put them on a belt and draw the ends together for making the joint.

A design of belt adjuster altogether different from the old-style clamp is made by the Cleveland Fabric Belting Co., 1473 West 110th St., Cleveland, Ohio, and is illustrated herewith.

The adjuster is made with two clamps that are quick-acting, one side of each having a shackle fastening. This



BELT ADJUSTER APPLIED TO BELT

enables the operator to put the clamp over a slack belt and swing on the shackle in short order. With the clamp tight on the belt, it is only necessary to turn a crank until the desired tension has been obtained by bringing the clamps nearer together.

The crank drives the rack gear by a worm gear. The adjuster can be easily separated in two sections for carrying from one job to another.

The adjuster is made of malleable iron in sizes to handle belts from 1- to 12-in. widths.

"Ko-Shovel" Stoking Machine

The stoking device shown in the illustration has been perfected recently by the Goetz Co., of Chicago. It is designed to inject coal into the furnace in small frequent charges. Its operation is continuous, and as the fire-doors remain closed, combustion is carried on at a uniform rate with no sudden changes in temperature. There is no inrush of cool air such as occurs in hand firing, which is detrimental to high efficiency and which also has its effect upon boiler upkeep. Consequently the design of the "Ko-Shovel" is conducive to high economy in the furnace. The machine handles successfully any grade of fuel up to 1½-in. bar screenings or down to No. 5 washed or unwashed coal. Alternate charges of ½ and 3/8 lb. of fuel are fired by the strokes of the plunger, the lighter charge being thrown with greater force. The total quantity is determined by the speed of the engine or motor driving the stoking machine.

It is customary to install the "Ko-Shovel" either directly in the upper part of the fire-door opening or

immediately above the door in the boiler front. The entering chute and the plunger require a space 5 in. deep. When the machine is installed in the fire-door opening, a new door is provided, so that it will not prevent or interfere in any way with firing by hand or with cleaning the fires through the fire-doors.

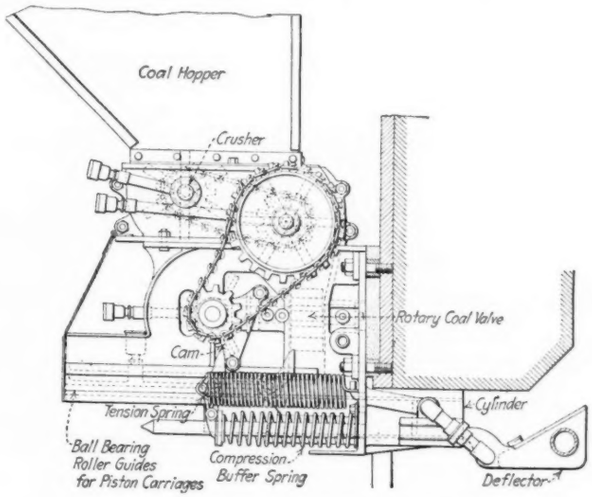


FIG. 1. DETAILS OF STOKER CONSTRUCTION

Reference to Fig. 1 will show the method of operation. Coal is fed into a hopper having sufficient capacity to run the boiler for about an hour. The amount of coal, or the number of charges entering the furnace is determined by the speed of the driving mechanism, which may be automatically controlled by a steam-pressure regulator. The coal comes down through a crusher below the hopper and through a rotating coal valve, operating at one-half the speed of the mainshaft, and drops upon the horizontal piston, which injects the coal into the fire. By the cams

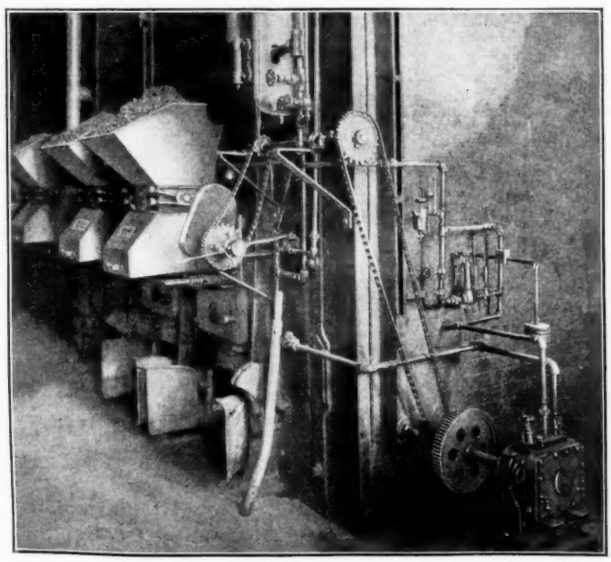


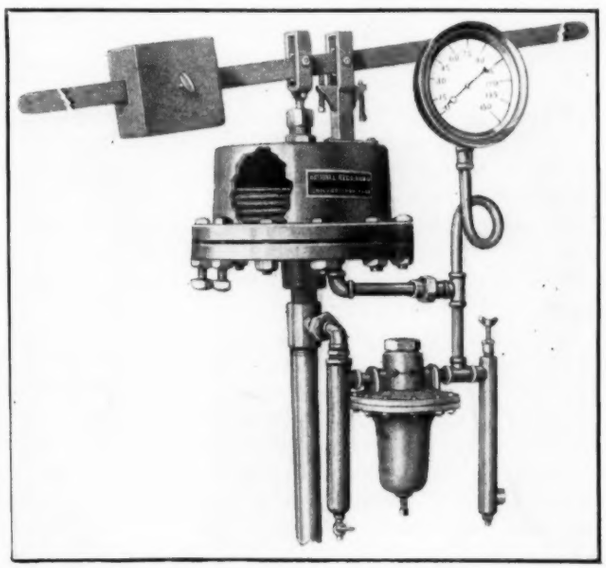
FIG. 2. TYPICAL STOKER INSTALLATION

shown in the drawing, the piston is pushed back to the end of its stroke. It will be noticed that there are two rollers on each cam and two lugs attached to the piston carriage for this purpose. After the piston is pulled back far enough, the coal drops into the cylinder, and as soon as the cams break contact with the lugs, it is discharged with considerable force into the furnace by a tension spring on each side of the piston. Compression springs attached to the stoking-machine body absorb the shock and bring the piston gradually to rest. Upon leaving the cylinder the coal strikes a deflector plate and is distributed uniformly over the grate.

From Fig. 2, which is a view of an installation made on Heine boilers, it is evident that chain and sprocket drive is employed throughout. Improved combustion due to the small and frequent charges of coal and the burning of the fine dust particles before they have an opportunity to fall through the grate or pack together and shut off the air supply, the exclusion of excess air, the comparative cheapness of the machine and its ready application to any existing boiler front are the principal claims advanced by the exclusive distributor of the device, George A. Kohout & Co., 403 Monadnock Building, Chicago, Ill.

Metaphram Damper Regulator

Recently the National Regulator Co., of Chicago, has perfected its "Metaphram" damper regulator for high-pressure work. As will be evident from the illustration, it consists of a plurality of separable collapsible corrugated metal disks connected by threaded studs and inclosed in a steam-tight casing. On top of the casing is mounted a fulcrum which supports the lever shown. To one end of this lever the damper is attached, and on the other end is a counterweight which balances the weight of the damper. The position of the latter is controlled by the diaphragm through a stem extending from the diaphragm through the top of the casing to the lever. Provision is made to mount the fulcrum on either side of the stem or to turn it about in either location and give a longer or shorter leverage as desired.



DAMPER REGULATOR READY TO CONNECT

Steam at boiler pressure enters the inside of the metal diaphragms through the central pipe leading into the bottom of the casing. Through a tee connection to the supply pipe steam also passes through a strainer and reducing valve into the casing of the regulator around the outside of the metal diaphragm. An expansion trap, shown at the right of the reducing valve, removes any

condensation. The pressure is thus equalized on both sides of the diaphragms.

When the boiler pressure rises to the point at which the reducing valve is set, the latter prevents further pressure from accumulating outside of the diaphragms. Any increase in steam pressure admitted to the inside of the diaphragms causes them to expand, forcing up the stem, operating the lever and gradually closing the damper. As the steam pressure drops toward the point at which the reducing valve is set, the lever gradually returns to its normal position. The regulator operates on a differential pressure that does not exceed 10 lb. In accordance with the steam pressure the damper is made to assume any position between that of full open and full closed. In other words, it does not swing from one limiting position to the other. The action is graduated so that the damper may occupy any intermediate position. This is a desirable feature in any damper regulator. The sensitiveness of the control depends largely upon the friction of the damper and on the position of the fulcrum.

A Salt-Water Pumping Plant

SYNOPSIS—The salt-water pumping station of the Standard Oil Co., El Segundo refinery, while embodying standard features throughout, has these combined in a manner worthy of notice.

El Segundo refinery of the Standard Oil Co. is on the crest of Aeolian Hills, along the Pacific Ocean near Redondo, Calif. The stills are approximately 150 ft. above the mean sea level. Salt water for condensing purposes is obtained from the ocean. The pump station is built on reinforced-concrete foundations and is a single-story reinforced-concrete building, 25x75 ft., with panel-constructed walls and steel-trussed roof. The pumproom has a clear height of 25 ft. and is spanned by a hand-power crane. Circular iron stairs lead to the 15x25-ft. transformer room, in a tower rising 36 ft. 10 in. above the main floor. Fifty-cycle three-phase current, purchased from the Southern California Edison Co., is brought into this room through tile entrances. After passing through an automatic circuit-breaker, it is reduced to 440 volts in a bank of three 200-kv.-a. single-phase transformers. Energy consumption is metered on the low-tension side, and a white marble distributing board is installed in one end of the transformer room.

The floor level of the pumproom is 7 ft. above mean high tide, and a breakwater of wooden piles and boulders (Fig. 1) is built around the structure for protection against high rolling storm waves. Outside and at one end of the pumphouse is a reinforced-concrete suction pit 25 ft. deep, 26 ft. long and 16 ft. wide, inside measurements. Water is drawn into this pit through a siphon made of 36-in. bell-and-spigot joint cast-iron pipe, with lead and hemp calking. Two vacuum towers 24 ft. high and 18 in. diameter, interconnected at the top, are installed to maintain a vacuum on the siphon. These are exhausted by two 12x14-in. single-cylinder wet-vacuum pumps, driven at 50 strokes per minute by 15-hp. motors.

There is always considerable sand drawn through the siphon, and at certain seasons of the year enormous quantities of kelp are discharged in the suction pit. The sand is partly eliminated by settling in the suction pit, but is a more or less constant source of annoyance in the condensers and sewers. An effective and ingenious kelp screen has been devised. A six-sided revolving screen rack (Fig. 2) extends across the pit. Screens 4x6 ft. are hinged along one edge to the rack. The screens on the under side of the roll hang vertically, and the bottom one seats against an angle iron embedded in the concrete of the pit floor. The pump suctions are in the rear of this angle iron, and the current draws the kelp against the screen, where it is caught by a number of prongs. When the screen becomes covered with kelp to a degree that interferes with operation, the rack is rotated, permitting a fresh screen to seat on the angle iron, and raising the kelp-covered one above the water surface, where the kelp is removed by forks and rakes. A hand-power crane spanning the pit is used to rotate the rack when it is too heavily loaded to move with pry bars.

The pumps are shown in Fig. 3. No. 1 is a four-stage turbine with a 12-in. suction and a 10-in. discharge. It

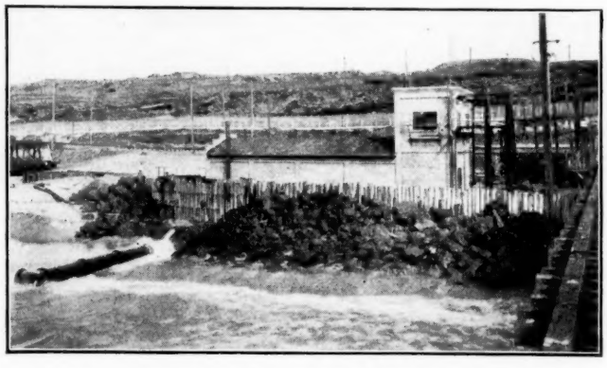


FIG. 1. REINFORCED-CONCRETE PUMP HOUSE

has a rated capacity of 3,000 gal. per min. against a head of 425 ft. when driven at 1,000 r.p.m. It is directly connected to a 500-hp. motor of the slip-ring type with hand control, set on the same sub-base. It discharges into a high-pressure main used for fire service only.

Unit No. 2 consists of a two-stage turbine pump with a 10-in. suction and discharge, and rated at 2,500 gal. per min. against a 181-ft. head when driven at 1,000 r.p.m. A 200-hp. squirrel-cage type motor is connected on the same sub-base. A spare pump unit is kept on the floor alongside, and in case of accident can be connected to the motor in a very short time.

Unit No. 3 consists of a single-stage volute pump with 12-in. discharge and 16-in. suction, rated at 4,500 gal. per min. against a head of 181 ft., when driven at 1,500 r.p.m. A 350-hp. motor of the squirrel-cage type is mounted on the same sub-base.

No. 4 unit is a single-stage volute pump with 10-in. suction and 6-in. discharge. It is rated at 2,100 gal. per min. when driven at 1,800 r.p.m. It is driven by a water turbine, which will be described.

Units 2, 3 and 4 discharge into a 36-in. cast-iron header, laid in an open concrete pit outside the building.

A venturi meter throat is connected in this header, while the indicating and recording gages are set inside the building. A close watch is kept on the venturi readings, and changes of efficiencies are promptly investigated. A number of serious accidents have thus been prevented.

From the header a 24-in. bell-and-spigot joint cast-iron pipe line leads to the still yard, where it is divided into a number of feeders leading to the surface condensers of each still. The discharge is caught in a concrete sewer system. The pickup sewers are 24 in. diameter and laid on a 0.2-per cent. grade. These feed into a 36-in. concrete sewer laid on a 0.1-per cent. grade, which carries the water to a large reinforced-concrete separator of the baffle-plate type, where the oil collected in the stills is removed.

A 48-in. tunnel laid on a 0.1-per cent. grade carries the water from the separator to a forebay, or penstock, built of reinforced concrete and set on the shore side of the sand hills at an elevation of 108.25 in. above mean sea level. A 24-in. bell-and-spigot joint cast-iron pipe line, 880 ft. long, is laid from the forebay to the pumping station; a short run of cast-iron pipe carries overflow water to a gully emptying into the sea. Because of the high grade

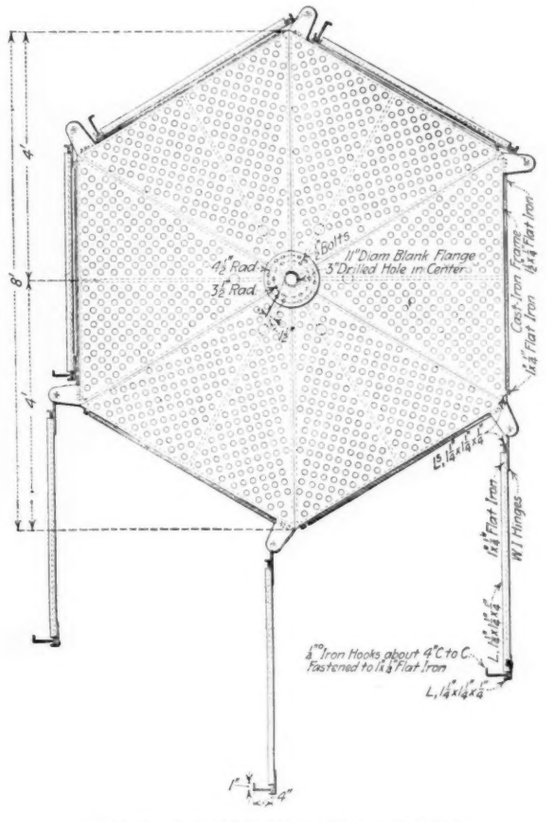


FIG. 2. REVOLVING KELP SCREEN

of the sewers and the low grade of the tunnel, when the water is drawn quickly out of the forebay, a peculiar wave action known as the "bore" takes place in the tunnel, and waves two feet or more in height will rush out of its mouth, causing the forebay to spill over.

Just outside the pumphouse a branch Y terminates the pipe line. The straightaway tapers through a reducer to 14 in. and flanges to a gate valve with its handle extended into the pumphouse. Beyond the gate a short run of cast-iron pipe permits the water to discharge into the ocean.

The branch of this Y turns on a 6-ft. radius and connects to a 24-in. hub gate valve. A reducer beyond this connects to a 14-in. 45-deg. elbow, passing through the power-house wall, and a 14-in. cast-iron line connects to the Pelton-Francis turbine. The waterwheel was designed especially for resisting the action of hot sea water. It is a double-discharge unit with hand-operated vanes

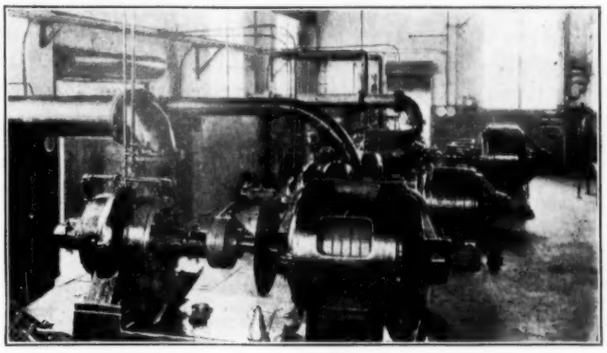


FIG. 3. INTERIOR OF PUMPROOM, SHOWING MOTOR-DRIVEN PUMPS

and develops 140 hp. The draft tubes are cast iron and are connected to a cast-iron pipe line extending into the ocean at 0 ft. elevation. It is mounted on the same base with a 6-in. pump, which it drives through speed-increasing gears of the herringbone type.

These gears, when first installed, heated badly. Many kinds of oil were tried, but the best results were obtained by using a dark summer lubricating oil of approximately 112 viscosity at 212 deg. F., a comparatively inexpensive grade. As a further safeguard against heating, an air-cooled system of pipes has been arranged, through which the oil from the gears is circulated.

The quantity of water required for condensing purposes varies widely, and it has been found that when the demand is only enough to require the operation of No. 2 unit, the plant efficiency is much better when the turbine is operated about twenty minutes, then shut down until the sewers, separator and forebay are filled, and the cycle repeated. When, however, the demand is great enough to require the operation of No. 3 unit, the turbine can be operated continuously, resulting in a clear saving of 140 hp. purchased energy, as the condenser water must be returned to the sea.

Factors of Evaporation

Temperature of Feed, Degrees	Boiler Pressure 30	40	50	60	70	80	90	100	120
32	1.207	1.211	1.214	1.217	1.220	1.223	1.225	1.227	1.231
40	1.199	1.203	1.206	1.209	1.212	1.214	1.217	1.219	1.223
50	1.188	1.192	1.196	1.199	1.201	1.204	1.206	1.208	1.212
60	1.178	1.182	1.185	1.188	1.191	1.194	1.196	1.198	1.202
70	1.167	1.171	1.175	1.178	1.180	1.183	1.185	1.188	1.192
80	1.157	1.161	1.165	1.168	1.170	1.173	1.175	1.177	1.181
90	1.146	1.151	1.154	1.157	1.160	1.162	1.165	1.165	1.171
100	1.136	1.140	1.144	1.147	1.150	1.151	1.152	1.156	1.160
110	1.126	1.130	1.133	1.136	1.139	1.142	1.144	1.146	1.150
120	1.116	1.120	1.123	1.126	1.129	1.131	1.134	1.136	1.139
130	1.105	1.109	1.113	1.116	1.118	1.121	1.123	1.125	1.129
140	1.095	1.099	1.102	1.105	1.109	1.110	1.113	1.115	1.118
150	1.085	1.088	1.092	1.095	1.098	1.100	1.102	1.104	1.107
160	1.074	1.078	1.081	1.084	1.088	1.090	1.091	1.094	1.097
170	1.064	1.067	1.071	1.074	1.077	1.079	1.081	1.084	1.087
180	1.053	1.057	1.060	1.064	1.067	1.069	1.071	1.073	1.077
190	1.043	1.047	1.050	1.053	1.056	1.058	1.061	1.063	1.064
200	1.032	1.036	1.040	1.043	1.047	1.049	1.051	1.052	1.056
210	1.022	1.026	1.029	1.032	1.035	1.037	1.040	1.042	1.045

Boiler performances are compared as though the feed water was 212 deg. F., and the steam at atmosphere pressure or 212 deg. F. To do this the table is handy.

Example—Feed water 120 deg. steam 80 lb., actual evaporation 7 lb. of water per pound of coal. To compare or give test rating, look in table opposite 120 deg. and under 80 and find 1.131, multiply 1.131 by 7 = 7.917, as the amount that would have been evaporated had the feed water been 212 deg. instead of 120.

Proportioning Cylinder Ratio in Compound Engines

By R. L. Wales

SYNOPSIS—A suggestion regarding the improvement in the present method of proportioning compound-engine cylinders. Also a visualization of the way in which the economy of the compound over the simple engine is effected.

The question of cylinder ratios for a compound engine is commonly determined from the point of view of equalizing the work between these cylinders. This is a mechanical point of view. In marine engines, where the flywheel effect is a minimum, it is important for smoothness of operation; in cross-compound stationary engines, where flywheel effects are possible, the matter becomes relatively less important; in tandem compounds, where the thrust from both pistons is received upon a common rod, the necessity for an equal distribution of work between the two cylinders becomes a minimum.

Compound Versus Simple Engines

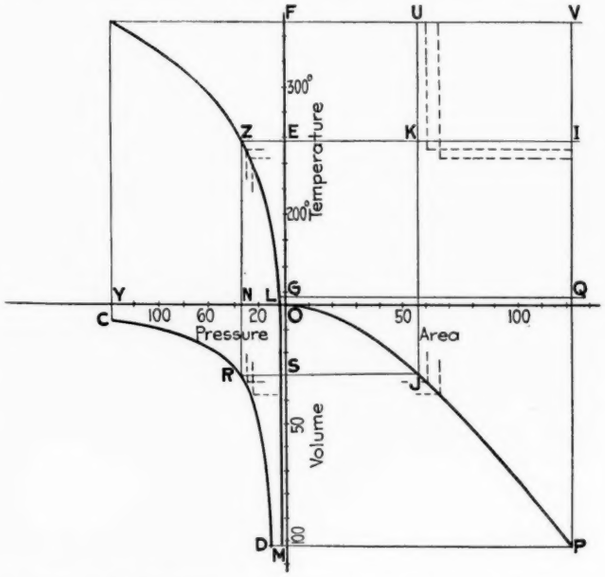
Although there are certain mechanical advantages of the compound over the simple engine, the fundamental reason for resorting to two cylinders rather than retaining the simpler form is a thermodynamic and not a mechanical one—namely, to cut down the losses due to initial condensation incident to the wide temperature range accompanying a large ratio of expansion. If the fundamental reason for compounding an engine is to effect a saving in initial condensation, it would appear logical to approach the design of such an engine with the object of making that saving a maximum, at least where the mechanical requirements are not preëminent.

The purpose of this article is to present a graphic method for showing the relative saving in initial condensation due to using a compound engine of any cylinder ratio instead of a simple engine working under the same terminal conditions, and for suggesting the proper ratio to make that saving a maximum. To make this plain the accompanying chart of temperature, volume, pressure and area is shown.

This method depends on the two following assumptions: That initial condensation is taken as proportional to the surface area on which condensation can take place, and also that it is proportional to the difference in temperature to which the surfaces are subjected. These assumptions are confessedly only approximate. In regard to the first it is apparent that one square inch of surface of the barrel of the cylinder would not be of the same effect in causing initial condensation as a square inch on the cylinder head, etc. The assumption is in error to this extent. In regard to the second, although heat transfer is in general proportional to the difference in temperature, here the temperature difference is not constant, but changing throughout the period of expansion. One authority asserts on empirical grounds that initial condensation is proportional to the square root of the temperature difference instead of to that difference. The effect of different points of cutoff is neglected in the following discussion.

Referring to the diagram, the upper left-hand quadrant is one of pressure and temperature. In this space is plotted a curve giving the relation between pressure and temperature for saturated steam as given in the steam tables. The lower left-hand quadrant gives pressure-volume relations as on the indicator diagram. The upper right shows temperatures and areas. The lower right shows areas and volumes.

The figure shows the application of the method to the 36x56x72-in. compound engine at New Bedford, Mass., a test of which is reported in Peabody's "Thermodynamics of the Steam Engine." Starting in the lower right-hand quadrant, the volume and surface area of the low-pressure cylinder are plotted at point *P*. Curve *OJP* indicates the surface area of a high-pressure cylinder of any size from zero up to that of the low-pressure cylinder. In



CURVES OF VOLUME, PRESSURE AND TEMPERATURE

the lower left-hand quadrant the volume of the steam at cutoff in the high-pressure cylinder is plotted at *C*. The curve *CD* is a rectangular hyperbola and represents the fall in pressure as the steam expands to the volume of the low-pressure cylinder. Clearance is neglected, also the receiver drop. The line *LM* shows the back pressure.

In this engine the ratio of the cylinders was such that the volume of the high-pressure cylinder is represented by the distance *OS*. The volume line *SR* cuts the expansion line at *R* and indicates that the pressure in the receiver was as shown by the projection on the pressure axis, or *ON*. Following the line of projection to the temperature-pressure curve at *Z* and projecting to the temperature axis determines that the temperature in the receiver was *OE*.

Similarly projecting from *C* it appears that the temperature at cutoff in the high-pressure cylinder was *OF*, and by projecting from the back-pressure line that the temperature at the exhaust from the low-pressure cylinder

was *OG*. The temperature range in the high-pressure cylinder is represented by the distance *EF* and that in the low-pressure cylinder by the distance *GE*.

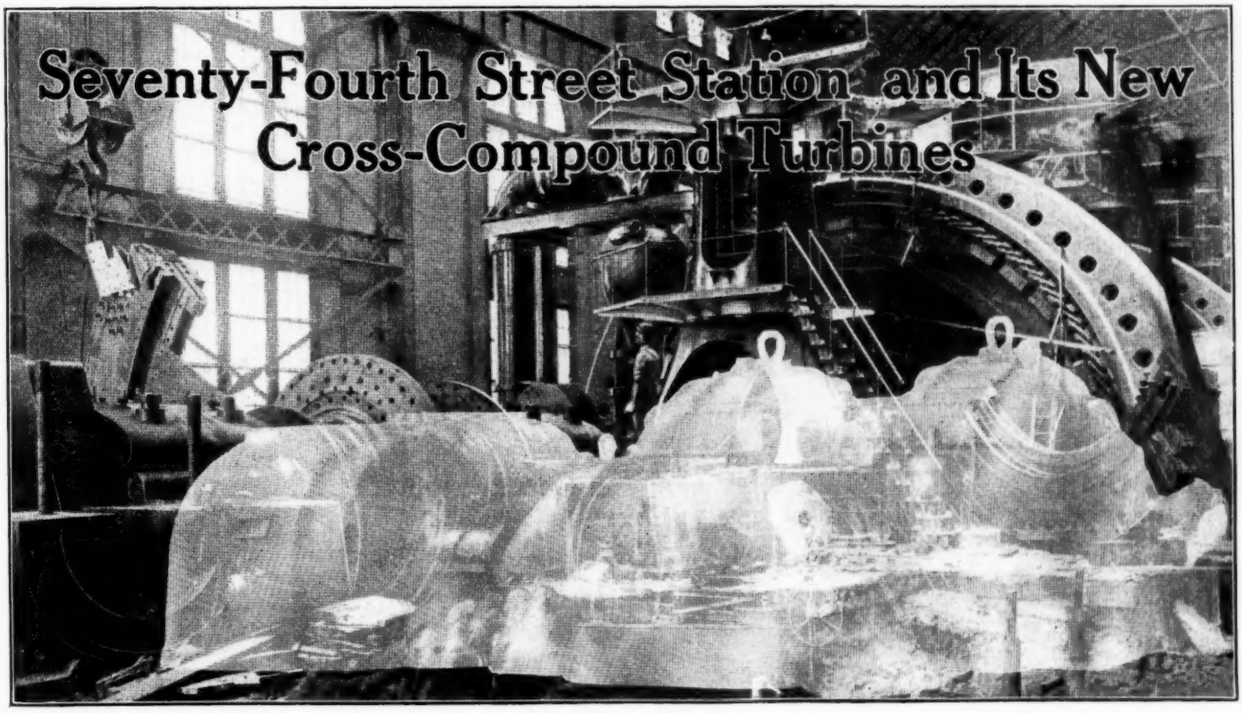
Since the volume of the high-pressure cylinder is *OS*, its surface area on which condensation can take place is *SJ* or *EK*. The surface area of the low-pressure cylinder has already been suggested as *GQ*.

In the rectangle *EFUK* the height measures the temperature range and the length the surface area for the high-pressure cylinder. The product of these dimensions, or the area of the rectangle, is according to the assumptions, a measure of the initial condensation in that cylinder. Similarly, area *EIQG* measures the condensation in the low-pressure cylinder. The sum of the areas represents the total condensation for the compound engine.

If the same total expansion and terminal conditions had been used in a simple engine, the total temperature range would have been experienced by that cylinder and would be shown by the full length *GF*. The surface dimension would be the same as for the low-pressure cylinder. The initial condensation would then be represented by the whole area *GFVQ*. The relative saving due to the partial expansion in a second cylinder is shown by the area of the rectangle *UVIK*.

In designing a new machine it should be the purpose to choose such a volume for the high-pressure cylinder that this area of saving should be a maximum. Running through the projections for various assumptions for this volume, as indicated in the dotted lines, is a ready means of determining just what volume gives this maximum area.

The foregoing is offered, not as a procedure that will in its exact present form lead the designer to infallible results, but rather as suggestive of a logical attack upon the problem which, with possible modifications of the assumptions, may lead to an improvement in the present method of proportioning compound engines. Its chief interest may lie in a better appreciation of the economy of the compound over the simple engine and a visualization of the way in which this increased economy is effected.



Seventy-Fourth Street Station and Its New Cross-Compound Turbines

By H. G. Stott*

SYNOPSIS—Remarks by Mr. Stott introductory to a paper by himself and W. S. Finlay, Jr.; entitled "Report of Efficiency Tests on a 30,000-Kw. Cross-Compound Steam Turbine," read recently before the annual meeting of the New York local section of the American Society of Mechanical Engineers. For paper and discussion of it see page 746 of this issue.

It may be of interest for me to make a few historical remarks. The station in which this apparatus is located was designed about the year 1900, so that from the time it was laid out until the present represents a period of about sixteen years. This installation illustrates very well the tremendous evolution of the art in that brief period. Sixteen years ago the turbine was hardly heard of in this country[1], a few practical experimental machines had been installed, and the large engines, which were then designed by the late Edwin Reynolds, of the Allis-Chalmers Co., of the double-compound type at the 74th St. plant, became known as the Manhattan-type engine.

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[1]Up to 1900 few, if any, turbines of capacities greater than 500 kw. had been designed. In that year a machine of 2,000 kw., the largest then built, was installed by the Hartford (Conn.) Electric Light Co. This machine created great interest not only because of its capacity, but because it was radical in having but one casing. Sir Charles Parsons at this time built a 1,000-kw. machine in which expansion was completed in two separate elements on a single shaft, each element having a casing of its own. There quickly followed a number of then large machines, some 16, in this country. The success of these machines is history. Turbine development then received an impetus that has carried it to the stage represented by the 74th St. machine.

That engine, as some may recall, consisted of a horizontal high-pressure connecting to a vertical low-pressure, at right angles on the same crankpin, and then on the other end, across from the generator, was a duplicate set, another horizontal high-pressure and another vertical low-pressure. Only two cranks were required for four cylinders. These engines established an enviable reputation for reliability. Their economy was as good as could be expected in reciprocating engines at that time, with comparatively moderate pressures of about 175 lb. and no superheat. The valves were the regular Corliss type.

It is interesting to recall what these units cost. They were 5,000-kw. rating, with a maximum overload capacity of 50 per cent., bringing the maximum load to 7,500 kw., which they would readily carry, but with correspondingly poor water rates. The water rate of the engine on test ran over 17 lb. per kw.-hr., and that was considered a very good record. The cost of these units—the engine, generator and condenser—was a little over $40 per kw. capacity, based on the rated capacity. To contrast that with what has been installed in place of them is rather interesting.

Comparative Costs of the Units

To go back just for a moment to the original plant, it was designed on what was practically a unit basis, as far as the engines and boilers are concerned; that is, there was a group of eight 520-hp. boilers per unit, and each unit could be operated independently of the rest of the plant, but normally was operated with all of the boilers connected together through two headers. The capacity, therefore, of boilers was a little over 4,000 boiler-horsepower for a 5,000-kw. unit.

In place of that equipment, the unit the test of which is described in the paper is a 30,000-kw. cross-compound high-pressure turbine, running at 1,500 r.p.m., exhausting into a low-pressure turbine running at 750 r.p.m. This unit of 30,000 kw. went between the same centers as the 5,000-kw. unit taken out. The cost of the modern unit including condensers was a little over $9, as compared with $40 for the engines.

The engines had an efficiency curve which took on approximately the shape of a V, so that it did not pay to run the units much over their rated capacity of 5,000 kw.

The large turbines which have been put in in place of the engines are operated by the same eight boilers; that is, the boiler capacity has not been increased except to put in underfeed stokers, so that we have simply changed a 5,000-kw. unit to a 30,000-kw. unit. That is about six to one in capacity, and on the same number of boilers. Of course this is being accomplished by corresponding low water rates, as shown in the tests described.

The improved water rate is obtained through the more economical turbine and through the use of the superheat, so that the actual water rate, not counting the superheat, is now with the present unit on an average of about 11.5 lb., as compared to about 17.5 on the average with the previous unit. This is a good illustration of the progress made in fifteen years in the art of developing steam motors.

The advantages that have been procured are: In the same floor space we get six times the capacity at a cost per kilowatt of approximately one-fifth of the original unit; and what is still better, the water rate, when reduced to saturated steam, has been reduced approximately 50 per cent.

The progress of the steam turbine has been so great that it today has practically displaced the gas engine. The thermal efficiency given in the tests described runs up to about 25 per cent. That is practically equal to the best gas-engine practice today. As the cost of the gas-engine unit is probably seven or eight times as great as the turbine, the gas engine has been practically put out of the running as far as large power-plant work is concerned.

Another interesting sidelight in this matter is the value of the turbine as a general prime mover in competition with anything else that could be cited. Fifteen years ago hydro-electric power developments were looked on as a choice investment, worth lots of money at almost any cost of development. Water powers were developed that cost $200, $250 and $300 a kilowatt. Today you could not get money for an investment of that kind. Why? Simply because the steam turbine has risen so high in efficiency and economy and decreased so much in first cost that it has driven out all possibility of developing many of these water powers. The reason is that when you allow for the fixed charges, the steam turbine can make power more cheaply than the high-priced hydro-electric development. That is another important point of view.

Take the case of Niagara Falls, where there is no dam required and where there is practically an unlimited supply of power. The steam-turbine plant can compete with Niagara Falls power today, as long as the load factor for the Falls power is less than 50 per cent. The only chance for financially successful water-power development is on high load factor.

Building Up the Voltage

By Gordon Fox

SYNOPSIS—A discussion of the causes usually responsible for failure of the voltage to build up, and for reversed polarity.

Whenever a self-excited direct-current generator is started from rest to be put into service, it becomes necessary to "build up" the armature voltage from zero to line value. Machines whose fields are excited from a separate source need not undergo this building-up process, since the rotation of the armature in the fully magnetized field causes the voltage to immediately assume line value or thereabouts. The building-up process is based on the fact that steel, once magnetized, retains part of its magnetism after the magnetizing force is removed. Consequently, when the armature of a generator is revolved in fields retaining a slight residual magnetism, a low voltage is generated in the armature conductors. This voltage is impressed on the shunt-field coils, causing a small current to flow through and strengthen the magnetism. This increase in field strength causes the armature voltage to increase, further strengthening the fields. Through this cumulative action the voltage rises, increment by increment, with increasing rapidity until the magnetism passes over the knee of the saturation curve and a position of stability is attained.

Failure of a generator to build up may be attributable to a number of causes, depending somewhat on whether the machine is being started for the first time or whether, after regular service, it fails without apparent cause. The former situation is the more common. Probably most of such occurrences are due to incorrect relation of armature and shunt-field connections. As previously stated, the field residual magnetism induces a small voltage which, being impressed on the shunt fields, starts to magnetize them. If this shunt-field action is of the same effect as the residual magnetism the generator builds up, but if this excitation opposes the residual magnetism the latter is neutralized and the machine is unable to build up in any direction.

One common cause of this condition is operation with opposite direction of rotation from that at which the machine was tested, without change of connections. This reversal of rotation changes the armature polarity, thus reversing the direction of current through the shunt fields so that they oppose the residual magnetism. If this condition is found to exist, the best procedure is to reverse the armature leads, or the armature and interpole together if interpoles are included. The shunt field alone should not be reversed if the generator is compound-wound. Such reversal would enable the generator to build up, but it would change the relation of shunt and series fields so that the compounding would be differential rather than cumulative.

Effect of Excessive Resistance

A second cause of failure to build up, and one which may occur at any time, is excessive resistance in the local circuit comprising the armature and shunt field. It is through this circuit that the initial small magnetizing current flows. Some generators will not build up if appreciable resistance is cut in at the field rheostat. Loose or poor contacts at any point sometimes cut off the flow of the initial magnetizing current. It must be borne in mind that the voltage generated in the armature due to field residual influence may be but 3 or 4 volts and insufficient to force current through any appreciable resistance. Poor brush contact is a fairly frequent cause of failure to build up the voltage. The "brush drop" is about 1 to 1.5 volts, and the two drops in series may add up to 2 or 3 volts. This value may be materially increased if a commutator is dirty or rough or if the brushes have extremely light tension. The voltage generated through residual magnetism must be great enough to overcome this brush drop.

Low-voltage generators are most likely to fail through insufficiency of initial voltage, 110-volt machines being less likely to build up than 220-volt machines. The residual magnetism is practically the same in either case, but the armature turns and terminal volts are less the lower the rated voltage. Generators for plating and similar work, rated at about 20 volts, frequently fail to build up at all. Such machines are therefore designed for separate excitation where there is a suitable source of power for that purpose.

As already mentioned, insufficiency of initial voltage may cause a generator to refuse to build up. As the generated voltage is proportional to the rotational speed, other conditions remaining unchanged, it is evident that if a generator is operated much below its rated speed, it may not give voltage.

Occasionally, after installing a generator and when starting it for the first time, it refuses to generate, no matter how carefully it is "nursed." Total loss or, at least, weakening of the residual magnetism can then be claimed. It must be said, however, that generally where loss of residual magnetism is claimed there are in reality troubles of one of the kinds mentioned. Loss of residual magnetism is not a common occurrence. It takes place sometimes in split-frame generators shipped "knocked-down."

Machines that have been subjected to extreme cold or to sudden knocks, such as in switching cars perhaps, are said to lose a part of their residual influence. Machines that are demagnetized require momentary excitation from a separate source to again establish the residual. This can best be applied by connecting the shunt fields across the station bus or the terminals of another generator. Where an isolated machine loses its residual, it becomes necessary to employ dry batteries or storage batteries for reëxcitation.

It sometimes happens that a generator builds up voltage of reversed polarity perhaps without apparent cause. This is an indication of reversed residual magnetism, which may be caused in several ways. If two generators are operating in parallel and the prime mover of one slows down and fails to carry load, its generator may become a motor, receiving power from the other unit and dragging its own engine. If this motored unit be disconnected under this condition, the polarity may be reversed because the series field, carrying a current in the opposite direction from normal, overpowers the shunt-field influence. A generator feeding storage batteries may have its polarity reversed from a similar cause.

Some engineers have a practice, not a good one, of letting an engine die down at quitting time without disconnecting the electrical load. If there happens to be on the line a fairly large high-speed motor connected to a load having inertia, trouble may ensue. This motor temporarily becomes a generator and pumps current back through the real generator which, through the action of its series fields, has its polarity reversed. The polarity in which a machine builds up is merely a question of the polarity of its residual magnetism. All that is necessary to reverse the generator polarity is to reëxcite the fields momentarily in the opposite direction from an external source.

Stresses Produced by Expansion and Contraction

Stress due to expansion and contraction may be approximately determined as follows, when the area in cross-section of the metal in the bar or pipe is known:

$$P = (T - t)\ ACE,$$

where

P = Thrust, or force, in pounds exerted;
T = Final temperature, deg. F.;
t = Initial temperature, deg. F.;
A = Area of metal cross-section;
C = Coefficient of expansion of the metal;
E = Modulus of elasticity of metal.

Example: The area of the metal in cross-section of a 6-in. extra strong pipe = 8.5 sq.in., so substituting values,

$P = (470 \text{ deg. F.} - 70) \times 8.5 \times 0.000007 \times 30{,}000{,}000 = 714{,}000$ lb. tension

The stress per square inch of area on the material in the pipe is $\frac{P}{A} = \frac{714{,}000}{8.5} = 84{,}000$ lb. per sq.in., but the area of metal at the root of the thread being less, the actual stress per square inch at that point would be proportionally greater.

Handy Tables for the Refrigeration Plant

PROPERTIES OF SATURATED AMMONIA

(Goodenough and Mosher; John Wiley & Sons, New York)

Pressure, Lb.	Temp., Deg. F.	Volume, Cu.Ft. per Lb.	Weight, Lb. per Cu.Ft.	Heat Content in B.t.u. of Liquid	Heat Content in B.t.u. of Vapor	Latent Heat in B.t.u. of Vaporization	Latent Heat in B.t.u. Internal
20	—15.9	13.45	0.0744	—50	534.0	584.3	534.7
21	—14.0	12.82	0.0780	—48.4	534.6	582.9	533.1
22	—12.2	12.27	0.0815	—46.5	535.1	581.5	531.6
23	—10.5	11.77	0.0850	—44.7	535.6	580.2	530.2
24	— 8.8	11.30	0.0885	—42.9	536.1	579.0	528.8
25	— 7.2	10.88	0.0919	—41.3	536.5	577.8	527.4
26	— 5.7	10.50	0.0953	—39.7	536.9	576.6	526.1
27	— 4.2	10.13	0.0987	—38.1	537.4	575.4	524.9
28	— 2.7	9.78	0.1022	—36.5	537.8	574.3	523.7
29	— 1.3	9.47	0.1056	—35.0	538.2	573.2	522.5
30	+ 0.1	9.17	0.1090	—33.6	538.5	572.1	521.3
31	1.4	8.90	0.1124	—32.2	538.9	571.1	520.2
32	2.7	8.64	0.1158	—30.8	539.3	570.1	519.1
33	4.0	8.39	0.1192	—29.5	539.6	569.1	518.0
34	5.3	8.15	0.1226	—28.2	540.0	568.1	516.9
35	6.5	7.93	0.1260	—26.9	540.3	567.1	515.8
36	7.7	7.73	0.1294	—25.6	540.6	566.2	514.8
37	8.9	7.52	0.1328	—24.4	540.9	565.3	513.8
38	10.0	7.34	0.1362	—23.2	541.2	564.4	512.8
39	11.1	7.16	0.1396	—22.0	541.5	563.5	511.9
40	12.2	6.99	0.1430	—20.8	541.8	562.6	511.0
180	89.4	1.666	0.6000	64.6	558.8	494.1	439.5
181	89.7	1.656	0.6034	65.0	558.8	493.8	439.2
182	90.1	1.647	0.6068	65.4	558.9	493.4	438.8
183	90.4	1.639	0.6102	65.8	558.9	493.1	438.5
184	90.7	1.630	0.6135	66.2	559.0	492.8	438.2
185	91.1	1.621	0.6168	66.6	559.1	492.4	437.8
186	91.4	1.613	0.6200	67.0	559.1	492.1	437.5
187	91.7	1.605	0.6223	67.4	559.2	491.8	437.2
188	92.1	1.596	0.6266	67.8	559.3	491.5	436.8
189	92.4	1.588	0.6298	68.2	559.3	491.2	436.5
190	92.7	1.580	0.6330	68.6	559.4	490.9	436.2
191	93.0	1.572	0.6362	68.9	559.4	490.5	435.9
192	93.4	1.563	0.6395	69.3	559.5	490.1	435.5
193	93.7	1.555	0.6428	69.7	559.6	489.8	435.2
194	94.0	1.548	0.6460	70.1	559.6	489.5	434.9
195	94.3	1.541	0.649	70.5	559.7	489.2	434.5
196	94.6	1.533	0.652	70.8	559.7	488.9	434.2
197	94.9	1.526	0.655	71.2	559.8	488.6	433.9
198	95.2	1.519	0.658	71.6	559.8	488.3	433.6
199	95.5	1.512	0.661	71.9	559.8	488.0	433.3
200	95.9	1.504	0.665	72.3	560.0	487.6	433.0
202	96.5	1.489	0.672	73.1	560.1	487.0	432.3
204	97.1	1.474	0.679	73.8	560.2	486.4	431.7
206	97.7	1.460	0.685	74.6	560.3	485.8	431.7
208	98.3	1.447	0.691	75.3	560.4	485.1	430.5
210	98.9	1.433	0.698	76.0	560.5	484.5	429.8
212	99.5	1.419	0.705	76.7	560.6	483.9	429.2
214	100.1	1.406	0.711	77.4	560.7	483.3	428.6
216	100.7	1.394	0.717	78.1	560.8	482.7	428.0
218	101.2	1.382	0.724	78.8	560.9	482.1	427.4
220	101.8	1.370	0.730	79.5	561.9	481.5	426.8

SOLUBILITY OF AMMONIA IN WATER AT DIFFERENT TEMPERATURES AND PRESSURES

(Sims)

Absolute Pressure, Lb. per Sq.In.	32 Deg. F. Lb. Ammonia per Lb. Water	32 Deg. F. Vol.	68 Deg. F. Lb. Ammonia per Lb. Water	68 Deg. F. Vol.	104 Deg. F. Lb. Ammonia per Lb. Water	104 Deg. F. Vol.	212 Deg. F. Lb. Ammonia per Lb. Water	212 Deg. F. Vol.
14.67	0.899	1,180	0.518	683	0.338	443	0.074	97
15.44	0.937	1,231	0.535	703	0.349	458	0.078	102
16.41	0.980	1,287	0.556	730	0.363	476	0.083	109
17.37	1.029	1,351	0.574	754	0.378	496	0.088	115
18.34	1.077	1,414	0.594	781	0.391	513	0.092	120
19.30	1.126	1,478	0.613	805	0.404	531	0.096	126
20.27	1.177	1,546	0.632	830	0.414	543	0.101	132
21.23	1.236	1,615	0.651	855	0.425	558	0.106	139
22.19	1.283	1,685	0.669	878	0.434	570	0.110	140
23.16	1.336	1,754	0.685	894	0.445	584	0.115	151
24.13	1.388	1,823	0.704	924	0.454	596	0.120	157
25.09	1.442	1,894	0.722	948	0.463	609	0.125	164
26.06	1.496	1,965	0.741	973	0.472	619	0.130	170
27.02	1.549	2,034	0.761	999	0.479	629	0.135	177
27.99	1.603	2,105	0.780	1,023	0.486	638	.....	....
28.95	1.656	2,175	0.801	1,052	0.493	647	.....	....
30.88	1.758	2,309	0.842	1,106	0.511	671	.....	....
32.81	1.861	2,444	0.881	1,157	0.530	696	.....	....
34.74	1.966	2,582	0.919	1,207	0.547	718	.....	....
36.67	2.020	2,718	0.955	1,254	0.565	742	.....	....
38.60	.....	....	0.992	1,302	0.579	764	.....	....

PROPERTIES OF SATURATED CARBON DIOXIDE

(Amagat)

Temperature, Deg. F.	Pressure, Absolute, Lb. p.r Sq.In.	Volume, Cu.Ft. per Lb.	Weight, Lb. per Cu.Ft.	Heat in B.t.u. per Lb. Above 32 Deg. F. of Liquid	Latent of Vapor	Total
—22	213.0	0.4323	2.32	—25.72	126.13	100.41
—13	248.5	0.3674	2.73	—21.87	122.67	100.80
— 4	288.3	0.3132	3.19	—17.87	118.86	100.99
+ 5	333.7	0.2674	3.74	—13.73	114.71	100.98
14	384.8	0.2286	4.37	— 9.38	110.12	100.74
23	440.2	0.1952	5.12	— 4.82	105.04	100.22
32	502.7	0.1669	6.00	0.00	99.34	99.34
41	573.3	0.1422	7.03	+ 5.17	92.91	98.08
50	648.9	0.1205	8.30	10.76	85.54	96.30
59	732.7	0.1010	9.90	17.01	76.84	93.85
68	825.0	0.0840	11.92	24.21	66.15	90.36
77	928.7	0.0672	14.85	33.19	51.91	85.10
86	1,038.0	0.0474	21.09	47.50	26.88	74.38
87.8	1,060.7	0.0412	24.27	53.77	15.04	68.81
88.4	1,069.3	0.0346	28.95	61.45	0.00	61.45

PROPERTIES OF SATURATED SULPHUR DIOXIDE

(Cailletet and Mathias)

Temperature, Deg. F.	Pressure, Absolute, Lb. per Sq.In.	Volume, Cu.Ft. per Lb.	Weight, Lb. per Cu.Ft.	Heat in B.t.u. per Lb. Above 32 Deg. F. of Liquid	Latent of Vapor	Total
—40	3.12	22.715	0.044	—21.33	178.56	157.23
—31	4.26	17.121	0.058	—18.83	177.36	158.53
—22	5.54	13.177	0.076	—16.29	175.99	159.70
—13	7.24	10.307	0.096	—13.72	174.42	160.70
— 4	9.23	8.223	0.122	—11.07	172.66	161.59
+ 5	11.79	6.668	0.150	— 8.39	170.68	162.29
14	14.77	5.290	0.189	— 5.65	168.48	162.83
23	18.32	4.328	0.231	— 2.84	166.08	163.24
32	22.44	3.574	0.280	0.00	163.48	163.48
41	27.40	2.950	0.340	+ 2.90	160.65	163.55
50	33.23	2.437	0.411	5.85	157.61	163.46
59	39.90	2.036	0.493	8.86	155.36	163.22
68	45.57	1.715	0.585	11.92	150.93	162.85
77	56.23	1.443	0.696	15.03	147.28	162.31
86	66.31	1.218	0.824	18.20	143.80	161.60
95	77.53	1.042	0.964	21.42	139.32	160.74
104	90.17	0.882	1.141	24.68	135.05	159.73

PROPERTIES OF CALCIUM CHLORIDE BRINE

Pure Salt Weight, Per Cent.	Degrees Baume, at 60 Deg. F.	Degrees Salinometer at 60 Deg. F.	Specific Gravity at 60 Deg. F.	Salt Lb. per Cu.Ft. of Solution	Specific Heat	Freezing Point, Deg. F.
3	3	12	1.024	3¼	0.980	+29
5	6	27	1.041	7½	0.964	+27
7	9	36	1.058	8½	0.936	+25
9	10	40	1.076	11¼	0.911	+23
10	11	44	1.085	13	0.896	+21
12	13	52	1.103	15	0.884	+18
14	15	62	1.121	17	0.868	+14
18	20	80	1.159	19	0.844	+ 4
20	22	88	1.179	22½	0.834	—1.5
22	24	95	1.199	26	0.817	— 8
24	26	104	1.219	30	0.799	—17
26	28	112	1.240	34	0.778	—27
28	30	120	1.261	37½	.....	—41
30	32	...	1.283	41	.....	—54

PROPERTIES OF SODIUM CHLORIDE BRINE

Pure Salt, Weight, Per Cent.	Degrees Baume at 60 Deg. F.	Degrees Salinometer at 60 Deg. F.	Specific Gravity at 60 Deg. F.	Salt, Lb. per Cu.Ft. of Solution	Specific Heat	Freezing Point, Deg. F.
1	1	4	1.007	0.63	0.992	+30.5
2	2	8	1.015	1.26	0.984	+29.3
2.5	3	10	1.019	1.58	0.980	+28.6
3	3.5	12	1.023	1.92	0.976	+27.8
3.5	4.0	14	1.026	2.24	0.972	+27.1
4	4.5	16	1.030	2.57	0.968	+26.6
5	5.5	20	1.037	3.24	0.960	+25.2
6	6.5	24	1.045	3.92	0.946	+23.9
7	7.6	28	1.053	4.63	0.932	+22.5
8	8.7	32	1.061	5.30	0.919	+21.2
9	9.7	36	1.068	6.00	0.905	+19.9
10	10.7	40	1.076	6.70	0.892	+18.7
12	12.6	48	1.091	8.20	0.874	+16.0
15	15.7	60	1.115	10.40	0.855	+12.2
20	20.4	80	1.155	14.40	0.829	+ 6.1
24	24	96	1.187	17.78	0.795	+ 1.2
25	25	100	1.196	18.68	0.783	+ 0.5
26	25.8	...	1.204	19.50	0.771	— 1.1

Editorials

The Triumph of the Turbine

The story of the development and reorganization of the plant of the Interborough Rapid Transit Co., as told on page 733, by its superintendent of motive power, Henry G. Stott, is an engineering wonder. A dozen years ago the contest between the turbine and the engine was on. The turbine had yet to demonstrate its practicability and reliability. The reciprocating engine-driven unit had reached the limit of its practicable size in the four-cylinder 5,000-kilowatt unit of the Manhattan type. So long as the turbine kept under this capacity, what little difference there might have been in the steam rates of the two types of prime mover was probably in favor of the high-class piston engine.

As capacities went up above 5,000 kilowatts, the turbine began to assert its superiority. As the problems of its design were solved, and its manufacture began to be standardized, the investment called for per kilowatt rapidly became less, while the consolidation of 40,000 to 50,000 horsepower in a single machine has greatly reduced the amount of labor and attendance required. The net result of the development in Mr. Stott's case has been six times the capacity installed upon the same floor space, operated by the same boilers at an installation cost of $9 per kilowatt of rated capacity for the turbine against $40 for the discarded engines and with less than 65 per cent. of the labor required for their operation.

Of course this is not all due to the turbine. There has been an enormous increase in the output of a given installation of boilers, in which express firing and the mechanical stoker have played a big part. When the Seventy-fourth Street Station was built, there were installed for each of the 5,000-kw. units eight boilers of 520 horsepower each. These boilers were designed for a working pressure of 225 lb., but were run at only 160 lb. for something over thirteen years. When the turbines were installed, a higher initial pressure was desirable, and after a thorough examination by the engineers of the station, the Police Inspection Department and the Boiler Insurance Co., the boilers were allowed an initial pressure of 215 lb., and even at that figure had a factor of safety of five. The only changes made were to replace the cast-iron headers by wrought-steel ones and to add superheaters. To those who appreciate the importance of the standing charges in the computation of the cost of power generation, these figures will be very significant.

Boiler-Room Instrument Boards

In the generating room of a power plant there are usually enough instruments to indicate and record conditions existing in the electrical circuits. In direct-current plants ammeters, voltmeters and wattmeters are regarded as essentials, and for alternating current a number of other instruments—frequency and power factor indicators, a synchronizer, etc.—are found on the modern switchboard. In fact, it is the custom to provide as many instruments as are needed to indicate in detail and to record operating conditions. The need for the instruments and their importance in economical operation are generally recognized, and the switchboard attendant is carefully instructed in their use and how to interpret the readings. All this is as it should be, but the generating room is only one department of the plant.

Instruments in the boiler room are of even greater importance, for the possibilities of saving are greater. It is only recently, however, that the need of accurate indications and records of boiler-room operating conditions has been appreciated. For years boiler plants have been operated with a steam gage as the only instrument. The efficiency obtained depended on the skill and judgment of the fireman, with no direct indications to help him form conclusions and compare present with past operation. Account was kept of the coal bills and less frequently of the amount of water evaporated. Daily or perhaps monthly totals showed whether results were above or below normal. If the latter, there was no indication to help locate the trouble and operation at low ebb must continue until the fireman stumbled onto the cause.

To operate boilers continuously at their highest efficiency the fireman must have instruments, and enough of them to keep him posted on all operating conditions tending toward this end. First of all, the fireman must know the steam pressure and the gage should be where it may be easily read at some distance from the boilers. With a varying demand, it is important to anticipate changes in load. Large-faced gages with a range of a few pounds in the vicinity of the working pressure are a great help, as changes in pressure of one-quarter pound are indicated and as soon as the tendency is evident, the fires may be prepared accordingly. Recording gages giving a continuous record of the pressure are also useful, as they show the fireman the results of his work as he is doing it. It goes without saying that the coal should be accurately weighed and the feed water measured. Steam-flow meters on each boiler are valuable, as they show the division of the load and in conjunction with the feed-water records indicate any loss of water by leakage or otherwise.

To maintain efficient combustion, gages to show the draft over the fire and the drop through the setting are essential, and a CO_2 outfit is a valuable acquisition. With these instruments it is possible to determine the best operating conditions, such as thickness of fuel bed and intensity of draft for various loads over the usual range of operation, and when these have once been determined, to maintain them, at the desired maximum.

It is evident then that the boiler room needs an instrument board. This must be conveniently located, and the firemen must be instructed in properly interpreting the readings. The latter requirement is important, as instruments neglected or improperly interpreted are of no assistance in bettering economy. Instruments first and then capable firemen should be the aim of every power-plant owner. The modern boiler room has no place for buskies. Intelligence is needed and must be had to obtain maximum economy.

"Taking Orders" from Advisory Officials

Increasing complexity in modern industrial organizations makes necessary a clear definition of responsibility for all executive actions. At the same time teamplay in the interests of the common employer is very important. Closely tied into these considerations is the question of taking orders from someone not one's immediate or direct superior.

What shall be the plant engineer's attitude toward a member of the company staff whose functions are advisory, although technical, when such an official seeks coöperation and very likely issues specific instructions? Here is a company, let us assume, with a number of departments, including an engineering staff concerned with development work and investigations, the preparation of designs and reports, and in short, occupied with multifarious duties of an advisory character. Then here is the plant engineer, in charge of operation and repairs of power equipment, reporting directly to the general superintendent and through him to the works manager. Shall the plant engineer receive and act upon instructions from the head of the engineering department direct?

Here are some questions capable of raising no little friction inside an organization unless they are foreseen by the management and clearly understood by all concerned. In theory, and often in practice too, it works out very well to issue instructions or to ask for service always through the plant engineer's immediate superior. Certainly such an official should invariably be advised as to any work requested of his subordinate, the plant engineer, no matter how apparently trifling it may be. Suppose the interests of the concern may require the running of a certain immediate test of equipment. Down comes the mechanical engineer of the company into the generating station, rushed with the responsibility of getting results at the earliest possible moment. The general superintendent may be out of reach at the time. Shall the plant engineer, then, refuse to take up the desired work on behalf of the mechanical engineer, an advisory official with no executive authority perhaps over him, or with so little as to make him feel as though someone was "butting in" with instructions not in accordance with routine procedure?

It is hard to answer such questions categorically, but on the whole, is it not the part of wisdom and real coöperation to dispose oneself and one's subordinates to help the mechanical expert to solve his problem? A fair way to start seems to be to express willingness to go ahead until otherwise instructed by one's immediate superior, at the same time taking such steps as may be possible to notify the superior that this or that work has been started. Of course such a beginning cannot safely be made if the work proposed involves the curtailment of service for whose quality and continuity the plant engineer is responsible. Often the work asked by the advisory member of the company staff involves nothing of this kind, and a refusal to start on the job until formally authorized may lose precious time and even fail to yield the necessary results in any way, on account of the transitory character of the conditions prevailing at the time the request for assistance is received.

All that any of the officials and employees should be seeking is the best good of the organization as a whole, and for this reason too close adherence to personal "rights" and too hazy conceptions of individual "authority" are inadvisable. The plant engineer should take no offense at being asked to coöperate in a piece of work by a member of the company staff whose functions are advisory, although good care should be taken to inform one's immediate superior what is afoot at the earliest opportunity. The establishment which works out these interrelations, however, in the shape of a clear-cut organization diagram and which thus defines the proper limits of authority and obedience, subject always to temporary changes due to emergency requirements, is bound to get better results than the one which fails to study the possible needs one department may have of another's services and the best and quickest ways of securing those services when occasion demands them.

It may be argued that with a clearly expressed set of rules or diagrams showing the limits of authority and responsibility, most of the internal friction hinted at will never appear; but there is a twilight zone in some companies between the authority of the executive and the powers of high advisory officials which to advantage may be illuminated by the searchlight of anticipation, with special reference to the relations of operating branches of the organization to officials of both classes.

Power-plant engineering is just one compromise after another.

Boiler Explosion on Ohio River Tug Kills Ten.—Headline. The authorities may now investigate why boilers explode.

The symptoms are so similar, we often wonder if a cause like that which makes a feed-water heater jump is responsible for hiccoughs.

Often we find ourselves dreaming of some picturesque Don who will educate managers to the importance of power-plant management.

When you feel discouraged about your slow progress to complete success, consider how long your friends who have "big" jobs were in getting them.

On the announcement of that seventy-thousand-kilowatt turbine Watt must have felt sure he really started something the day he contemplated that steaming kettle.

Sometimes we have dreams, and one of them that persists is that a future is ahead when there will be no engineers who do not thoroughly understand efficiency and practice it.

We have received numerous pleasant expressions of appreciation of the "Handy Hints and Clever Kinks for the Power Plant," on pages 660 and 661 of the May 9 issue. We expect to publish others from time to time and shall be glad to receive and pay for suggestions in this line.

Correspondence

Water-works Type Condensers Compared with Water-Tube

Referring to the article on page 178 of *Power* of Feb. 8—"Friction Loss Through Check Valve and Condenser"—which referred to a water-works type condenser serving a steam-turbine pumping engine, I have calculated the power to pass the water through the condenser alone and find that this is 19.7 water horsepower, or assuming a pump efficiency of 80 per cent., approximately 25 b.hp.

The same power expended in a separate circulating pump to force condensing water through the tubes of a water-tube type condenser, as used in electric power plants, would produce a much higher vacuum and correspondingly higher pump duty.

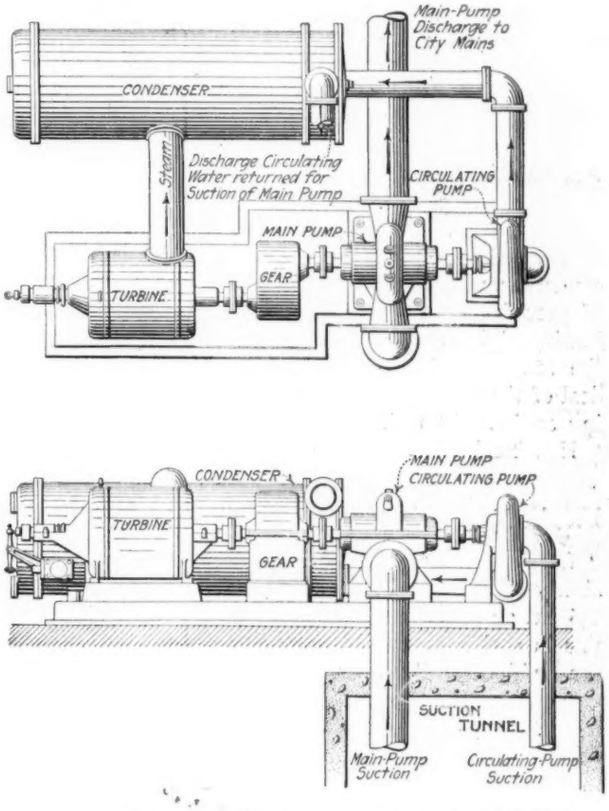
As stated in a recent paper before the American Society of Mechanical Engineers,[1] the so-called water-works condenser, in which condensation takes place inside of the tubes, is an excellent example of incorrect design for high vacuum and high rates of heat transmission, which is the more deplorable in a water-pumping plant because of the easily obtainable high vacuum. To obtain high coefficients of heat transmission, there must be rapid film agitation on the water side of the tubes, produced by high water velocity, and there must be low steam velocities and ample flow area on the steam side of the tubes, so as to insure steam penetration to all of the surface. In the water-works condenser the water is passed over the tubes at low velocity and the steam through the tubes at high velocity—just the reverse of what it should be.

In the case of the present unit it was stated in *Power* of Dec. 14, 1915, that the cooling water for six months of the year was below 40 deg. The acceptance test given in that issue and referred to in the issue of Feb. 8, 1916, shows a vacuum of 28.47 in. corresponding to 92 deg. exhaust temperature, with cooling water entering the condenser at 35 deg. and leaving at 38 deg. The excessive difference between the steam and leaving-water temperature is caused by the low efficiency of the surface, due in turn, as mentioned, to the lack of rapid film agitation on the water side and, second, to the fact that the steam never reaches a large portion of the surface, the area for the flow of steam into the tubes being so small that it cannot assume the very high specific volumes corresponding to the high vacuums and at the same time penetrate well into the condenser.

Reference to the acceptance-test figures shows that, even assuming the water rate of the main turbine and auxiliaries as steam condensed and assuming 1,000 B.t.u. per lb., the heat transmission per square foot of surface per hour per degree difference in temperature is only 126 B.t.u. If all of the surface were active—that is, if sufficient flow area were allowed and sufficient air-pump capacity provided to enable the steam to penetrate to all of the surface—this coefficient would have been several times as large and the steam temperature would have been brought down much closer to the water temperature.

If at the same time—that is, with steam reaching all of the condensing surface—the water-film agitation on the other side of the surface were sufficiently vigorous, the coefficient of heat transmission would be five to ten times as great.

With a water-tube condenser, as used in electric plants, the surface is made efficient in heat transmission on the water side by passing the water through the tubes at a high velocity and on the steam side by the proper spacing out of the tubes.



SUGGESTED ARRANGEMENT OF WATER-WORKS CONDENSERS

The available power in the present example is 25 b.hp., and assuming that a small independent circulating pump has an efficiency of 70 per cent. (which is obtainable), there would be available 17½ water horsepower, which would deliver 4,000 gal. per min. against a total head of 17 ft. This is ample for the friction in the piping and through the condenser, with water flowing through the tubes at approximately 5 ft. velocity.

On a basis of the same quantity of steam as before, this gives a temperature rise of the water of 7½ deg., which is comparatively low. The quantity of circulating water could be reduced to 3,000 or 2,000 gal. without

[1] "Performance and Design of High-Vacuum Surface Condensers," A. S. M. E., December, 1915.

occasioning excessive rise in temperature of the cooling water. The increase in efficiency of heat transfer on the water side of the tube alone would result in a coefficient of about 200, and basing the calculations on 35-deg. water and 42½-deg. outlet water and the same amount of condensing surface, the steam temperature obtained is 74 deg., which corresponds to a vacuum of 29.15 in. The vacuum of 29.15 in. would result in a gain of 7 or 8 per cent. in turbine economy and pumping duty.

This vacuum is moderate, considering the very low temperature of the water. By proper design of the condenser shell and spacing of the tubes so as to insure penetration of the steam and full efficiency on the steam side of the tube, this coefficient and the resulting vacuum would be still further increased. Under the conditions of 35-deg. condensing water, a vacuum of 29.5 to 29.6 in. should be maintained.

The layout of the sketch shows one arrangement in which the condenser is on the same level as the turbine and the circulating pump driven from the mainshaft.

With the turbine water-works pump, high vacuum is even more important than with the turbo-generator, because the load is constant and the load factor high, water is abundant and the duty, except with very high vacuum, considerably less than the better performances of reciprocating pumps.

The water-works type of condenser is just as obsolete as the reciprocating pump. The choice of this type of condenser, which no doubt has been made largely because of precedent, may simplify the plant slightly by eliminating piping and an extra pump, but it unduly hampers the condenser designer, upon whom, after all, a great deal of the continued success of the steam-turbine pumping engine will rest. PAUL A. BANCEL.

New York City.

Boiler Tubes Quickly Replaced

In the Apr. 4 issue, A. Arnois seems to doubt the ability of Edward Horsfield to replace a boiler tube in 4½ hr. and get the boiler under full steam back on the line in that time, as told by him in the Feb. 8 issue. It can be done easily by men of experience who have the proper tools to work with. I have often, during the noon hour, when burning clinkery coal, cleaned the fire, pushed the live coal back to the bridge-wall and banked it lightly, then throwing open all doors and dampers, have been able to enter the firebox and cut the slag out of the slots in the grate bars after waiting about 25 min. for them to cool; the boiler was steaming again after an hour.

Of course it is hot work, but it can be done. I do not mean that I went all over every grate bar; just the worst ones. There is plenty of cool air going through the firebox by this method. By cleaning the ashpit and filling it with cold water, the cooling process can be hurried. I believe that in an hour any furnace can be cooled in this way sufficiently to allow working in it. I have also entered the combustion chamber after an hour and rerolled a leaking tube end with the boiler under steam.

Removing a tube inside of 50 min. is often done with the aid of an inside-tube clamp and chain blocks, by keeping a steady strain on the tube after it is started and having a man hammer on the exposed end to loosen the scale. A tube can be inserted and rolled in 40 min. by using two expanders, and the tube may be beaded while the boiler is filling. The top manhole can be opened and a hose inserted in addition to water entering by the feed line. As soon as water covers the top tubes, the fire may be started, and by using wood, steam is generated in a short time. Allowing 1 hr. for cooling and 1 hr. 30 min. for removing and renewing the tube leaves 2 hr. for filling and steaming. This for a return-tubular boiler 16 ft. by 60 in. Although Mr. Arnois asks for an explanation from Mr. Horsfield, I thought he would like to hear from another. W. P. HUMPHREY.

Acra, N. Y.

[Rolling tubes under pressure is not to be encouraged.—Editor.]

Friction in Riveted Joints

In the issue of May 2, a letter from Joseph Goder on the subject of friction in riveted joints appears, and on this you comment editorially. As you say, this subject crops up occasionally, and I fully agree with you that the effect of the friction in riveted joints had better be forgotten, and let it merely do what good it can.

I am not familiar with the experiments of Prof. Carl von Bach, but I believe that no engineer would dare to give any credit to the friction of riveted joints. It can be imagined that hot rivets shrink away from holes as they cool, but it is one of those infinitesimal matters that are of no importance in practice. The behavior of good riveted joints in boilers proves this, for when such joints are calked they seldom leak, and they never rupture in consequence of slipping.

It was not necessary to wait for work executed by Professor von Bach 15 years ago, for many years before this a good deal was done on this subject in England, and in a book by Prof. Bindon B. Stoney, of Trinity College, Dublin, entitled "The Strength and Proportions of Riveted Joints," published 31 years ago by E. & F. N. Spon, several pages are devoted to the matter. I suggest that interested readers look up pages 35 to 38 and 74 to 77 of this book. F. W. DEAN.

Boston, Mass.

Graphite Cylinder Lubrication

It has been determined in practice that graphite is a superior lubricant, but owing to its being a solid, until recently no adequate means had been devised for properly feeding it to engine cylinders. Dry graphite is not as well suited for bearings and especially for engine cylinders, as when it is mixed with oil.

In experimenting with graphite and a fairly good grade of oil for cylinder lubrication for a period of two years under various conditions, a 26&48x48-in. Corliss engine, running 90 r.p.m., was used, the load varying from 800 to 1,200 hp.

In determining the amount of oil to be used, the horsepower was disregarded and the number of square feet covered by the piston was taken instead. It was found that the quantity of oil required under the various loads did not vary appreciably. The oil was cut down until the engine commenced to show signs of distress, then the quantity would be increased just enough to relieve it.

After a number of tests the quantity of oil was determined at which the engine could be run for a period

of 54 hr. This was approximately 4,500,000 sq.ft. per gal. The engine was operated under these conditions for 10 weeks. The cylinders were opened at intervals for inspection. Lubrication was invisible to the naked eye. It could be detected, however, with the aid of a microscope.

It was decided to try graphite and oil as a means of reducing the cost of cylinder lubrication, with the result that approximately 12,000,000 sq.ft. was lubricated per gallon of oil and graphite mixed. As in the first case, the cylinders were opened at intervals for inspection. The walls were shown to be in a better condition in every way, lubrication being plainly visible to the naked eye.

The proportion of graphite used is from 3 to 4 per cent. by weight. With this quantity there is no heavy deposit of graphite in the exhaust ports, receivers, exhaust pipe, piston rings, packing or valve springs.

Manville, R. I. JOHN L. ALLEN.

Convenient Oil-Groove Cutter

A handy little tool for forming oil grooves in long bearings of small diameter, such as motor bearings, is shown

TOOL FOR CUTTING OIL GROOVES

in the illustration. An old round file lends itself very readily to this use, if the body is annealed to prevent its breaking.

L. R. HEARVEY.

Washington, D. C.

When Is a Chimney Not a Stack?

I recently prepared specifications for two 10x200-ft. self-supporting steel chimneys. Upon opening the bids it was found that they were termed "steel stacks." In taking bids under a former specification for a masonry chimney, this did not occur. Upon looking the matter up it was found that a means or a structure for carrying off smoke could be termed a pipe, tube, funnel, shaft, conduit, flue, breeching, uptake, stack or chimney. It was also found that an accepted distinction among builders is, that if constructed of steel plates it is a "stack" and if of reinforced concrete or masonry it is a "chimney." This is not borne out to any great extent by writers dealing with the subject; they for the most part classify all under the name "chimney." Following are some of the uses of the terms met with in current literature and commercial practice:

Webster's New International Dictionary defines a chimney as an upright tube or flue of brick or stone to carry off smoke.

Taylor & Thompson, American Portland Cement Manufacturers' Association's Bulletin, General Concrete Construction Co., Heine Chimney Co., Weber Chimney Co., M. W. Kellogg Co., Bergen & Lindeman, Inc., Custodis Chimney Construction Co., H. R. Heinicke and Molesworth's Pocketbook employ the term "radial brick chimneys."

Kent and Gebhardt refer to brick, concrete and steel chimneys. John D. Adams, in *Engineering News*, writes of "The Design of Self-Supporting Steel Chimneys."

Riter-Conley, Lippett & Wood, E. Keeler Co. and Graver Tank Works say "steel stacks" in their advertisements.

An engineer depends to some extent for his knowledge of certain matters upon the literature and advertisements issued by manufacturers, and for this reason every effort should be made by them, as well as others, to be consistent.

THOMAS R. TARN.

Pittsburgh, Penn.

Wants Boiler-Cleaning Costs

I would like to have some information as to the cost of boiler cleaning in other engineers' plants. The following cost figures apply to one of our boilers. A certain number of variables enter into the problem, such as the character and thickness of scale, but these can be taken into account and comparison made on the basis of the cost per tube. The boiler is a B. & W. operated at 200 lb. pressure. The tubes were cleaned with a water turbine.

Size of tubes, in.	4
Number of tubes	287
Boring time, hr.	13½
Approximate thickness of scale, in.	3/32
Cost Data:	
Cutter wheels	$3.00
Water, 9,000 cu.ft., $0.18 per 1,000	1.62
Labor	8.78
Steam, 4,942 lb., $0.125 per 1,000	.62
Oil	.60
Total	$14.62
Cost per tube	$0.051

The number of pounds of steam is figured on displacement, with 10 per cent. allowed for condensation losses, from the number of strokes taken to clean each tube, the pump being operated on full boiler pressure.

Middletown, N. Y. HARRY A. COX.

Boiler Ruined by Valve in Water Column Connection

To show how easy it is, under some circumstances, to get caught with low water and ruin a boiler, the following is related. The plant contained a 75-hp. internally fired boiler, a 40-hp. engine and generator. When I arrived in the morning, there would be found a good fire and usually three gages of water.

One morning I tried the gage-cocks and got water from all three. Supposing the water was just a little high, I increased the draft, oiled up and started the engine. I heard something snap, followed by a roaring, hissing sound. I saw steam coming from every part of the boiler. I dumped the fire quickly and groped my way to more comfortable quarters, where I spent half an hour trying to answer embarrassing questions. After the excitement was over, the fusible plug was found blown and there were three rows of leaky tubes—some bent in the middle and others expanded so that they projected an inch beyond the flue-sheet. The boiler was repaired, but a new one was required about a year after.

Subsequent investigation showed how it happened and partly exonerated me. The boiler was equipped with a water column that could be shut off on the steam side. The column carried a whistle that would screech for low water, but would be silent for high water. It developed that during the night previous to the accident the steam pressure had gone too low for the injector to be started, and the night man, to stop the noise of the whistle, simply shut the valve above the water column. This foolish act caused the water to be pushed to the top of the glass and stopped the whistle from blowing, although the water was really below the bottom of the glass in the boiler. I, being young and inexperienced, took it for granted that if I got water from the gage-cocks, there must be water in the boiler. This shows the danger of having valves in the water-column connections CHARLES HAEUSSER.

Albany, N. Y.

Cracked Piston Head

The numerous discussions on cracked pistons and heads, which have been appearing recently, bring to mind an experience I had while testing a two-stroke-cycle high-compression oil engine having oil-cooled pistons. We had been taking the oil consumption every day, and no signs of trouble were visible. It so happened, however, that the engine was allowed to stand idle for a little over a week, and when turning it over previous to starting again, I noticed that there seemed to be considerable oil in the cylinder.

We removed the piston, and a crack in the head showed where the trouble was. An investigation showed a slight flaw in the metal, but it also showed something more interesting. The metal along the fractured section did not have the sharp appearance of newly broken iron, but rather it seemed to have been smoothed off slightly. Our explanation of this was that the crack had doubtless been in the head for some time, but when the engine heated up, the expansion in the metal of the head was sufficient to completely close the crack and thereby prevent loss of oil. This expansion and contraction of the metal would also account for the worn appearance of the iron at the fractured section.

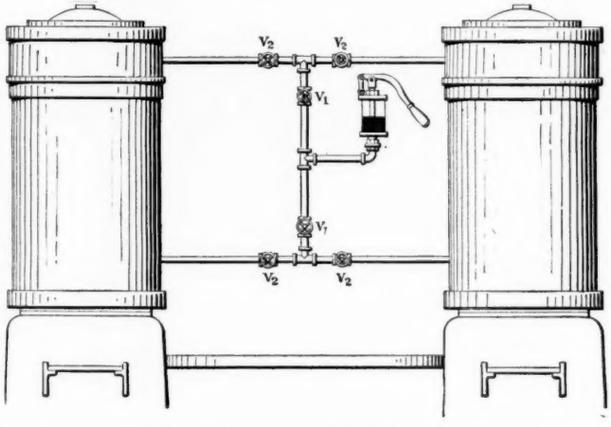
Erie, Penn. THEODORE M. ROBIE.

Reduced Compressor Capacity

Considerable trouble had been experienced with the oiling system of a 75-ton double-acting vertical twin-cylinder type ammonia compressor. A ½-in. line from the oil trap to the ammonia cylinders circulated the oil over and over again, and consequently bits of packing and dirt accumulated in it, clogging the oil lines frequently. During the winter, while the machine was being overhauled, the engineer decided to change the oiling system by taking out the old piping and piping it up as shown in the illustration, and also to connect in a small hand-feed oil pump. Having had so much trouble with the ½-in. oil lines, he decided to increase the size to ¾-in.

When the machine was started, it would not come anywhere near up to its normal capacity. As everything had been thoroughly overhauled during the winter it was expected to do much better than before, but the back pressure was much higher than usual and the head pressure much lower at the same speed and with the same amount of water going over the condenser. The machine was opened and thoroughly inspected, but everything was in first-class condition, so the manufacturers were appealed to and a man was sent from the factory to locate the trouble.

He first noticed that there was a peculiar intermittent hissing sound. The machine was shut down and the valves were taken out, but they were in first-class condition. While working on the valves, the factory engineer happened to notice the oiling system, and here was the cause of the trouble. The cranks were set 180 deg. apart, consequently when one cylinder was making the compression stroke the other cylinder on the same end was making the suction stroke. By referring to the illustration it will be seen that the gas would blow through the ¾-in. oil line fron one cylinder to the other, consequently



CYLINDER OILING SYSTEM PIPING

the amount discharged per stroke was considerably diminished, reducing the capacity of the machine. The two valves V_1 were taken from where they were and four valves V_2 put in, as shown. In the first instance the engineer had only two valves and did not want to ask for new ones, so he put them in as shown at V_1, not thinking of the consequences. After the change the machine ran all right. THOMAS G. THURSTON.

Chicago, Ill.

The Value of Friction Clutches

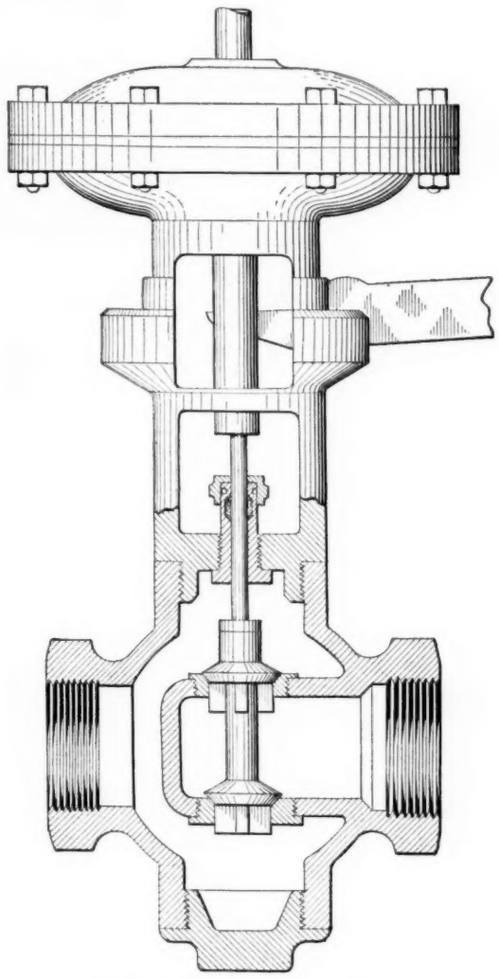
Friction clutches are not costly compared with the saving in coal resulting from their liberal use in shops and factories not operating at full capacity constantly. Several cutoff clutches may sometimes be advantageously placed on a single main lineshaft so that a whole department or section of a department can be thrown off or on at any moment, and every friction clutch is a "safety device," as any machine can be shut off in a moment.

Machines running idle often consume almost as much power as when in use, hence the saving where these machines can be cut off. The common practice is to cut out idle machines by throwing off the connecting belts, but this does not cut off that idle part of the lineshaft, where it happens to be one of the ends. (Obviously, the middle of a lineshaft cannot be cut out.) And to start the idle machines again the belt is put on a rapidly revolving pulley. This is an unsafe practice and is hard on the belt. S. F. WILSON.

New York City.

Pump Governor Experience

Some time ago we had trouble with the water cylinder and discharge-chamber gaskets and difficulty in keeping the pistons on our boiler-feed pumps packed. A different grade of packing was tried with no better results. A pressure gage was connected into the discharge line and showed the cause to be high pressure. Although the boiler pressure is only 100 lb., this gage indicated 260 lb. The pump governor was examined and the seats and disk were found to be badly cut, especially the lower seat. It was partly ground in with emery and the valve was reassembled, but the operation still was unsatisfactory. The



BALANCED TYPE VALVE USED

governor was again taken apart and a reseating machine used to true the seats until all the cuts were out. To maintain the same distance between the two seats so that the valve would fit, a tram was made to fit the seats before and after reseating to give the same distance between seats as before.

The valve was then placed in the lathe and turned down to fit the same tram, but when replaced, it did not rest on the lower seat, probably because of inaccuracy in tramming the seats, as the valve is small. Several light cuts were taken, and the valve was then ground in with a mixture of grease and powdered glass. This is better for grinding brass because emery scores and does not grind the seat smooth, and grease is better than oil as it does not run off. Both seats were ground at the same time, insuring a tight fit. After assembling and adjusting the governor, the gage in the feed line showed 110 to 115 lb. pressure, and the pump stopped when the boiler-feed regulators all closed or when the pressure went above 120 lb. No further trouble was experienced with gaskets or packing not holding. Since that time a gage has been permanently attached to the feed line.

A safety valve is often put on the feed line, and it is a good thing when set about 50 lb. above the boiler pressure, to relieve the line if the governor gets stuck or fails to close tightly when all the regulators close during the light-load period. Water relief valves, however, are not very reliable, because dirt and scale will form in them if not frequently used, causing them to stick or fail to seat tightly.

J. C. Hawkins.

Hyattsville, Md.

Power-Plant Losses at High Altitudes

In the issue of Apr. 11, page 515, in the article by A. V. Clarke, entitled "Power-Plant Losses at High Altitudes," there are some statements that are not borne out by my experience. The chart shown in Fig. 1 does not check with my figures. The elevation of my plant is 3,600 ft. above sea level. A barometer of the vernier-type mercury column is read at the beginning of each shift, or three times daily. The average of all readings from May 1, 1915, to date is 26.528 in. Allowing 0.491 for the weight of a cubic inch of mercury at 60 deg., this corresponds to a pressure of 13.025 lb. per sq.in., which does not check with Mr. Clarke's Fig. 1.

His statement, "To obtain the same evaporation from a boiler at a higher altitude as can be obtained at sea level, it is necessary to use forced draft, but the employment of this might increase the velocity of the air and gases through the grate and flues to such an extent that the operation of the boiler would be uneconomical and it would be better to have a larger one," is to my mind open to question.

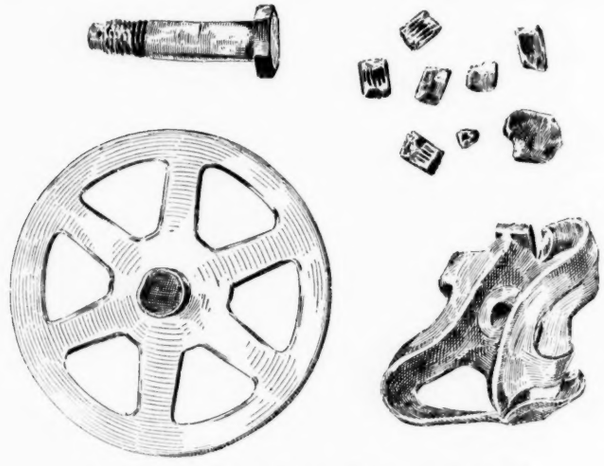
At this plant no forced draft is employed. The stack is 150 ft. high and maintains a draft at its base sufficient to take care of the maximum rating that the boiler plant has yet carried—about 140 per cent. The average combined efficiency of boilers and economizers for the seven months from September, 1915, to March, 1916, inclusive, is 84.3 per cent. This value has been corrected for the steam employed to atomize the fuel oil. This figure will, I believe, compare favorably with any plant located at sea level.

With regard to the engine alone I fail to see any great advantage accruing to the condensing type at the higher altitude. It is true that should a plant at sea level and one at, say, 3,600 ft. elevation use the same gage pressure and steam pressure, the latter will have a slight advantage, since steam at 200 lb. absolute and 500 deg. contains fewer heat units per pound than steam at 198 lb. absolute and 500 deg. Neglecting this, the vacuum is the determining factor. Since 24.5 in. of vacuum with a barometer of 26.5 in. will give the same absolute back pressure, corresponding temperature and heat content as a vacuum of 28 in. with 30 in. barometer, there is no advantage to be derived from this source.

Miami, Ariz. Charles Watt Smith.

Condenser Pump Accident

The broken and disfigured parts shown in the illustration were taken from a bucket condenser pump. The valve came loose early in the afternoon, but the engine had to be continued in service for the rest of the day,



BATTERED VALVE FROM VACUUM PUMP

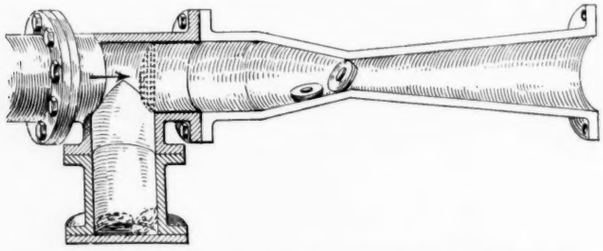
although the vacuum was impaired and forced injection had to be used.

On opening the pump that night, the spider and other parts were found in the condition shown, having been caught between the bucket and head plate, bent and battered, and during the process the pump gave an occasional hard thump when the loose parts got in the channelway. It is fortunate no more serious damage was done.

Methuen, Mass. W. E. Taylor.

Pump Valves Wedged in Meter

A venturi meter connected in the line from the house pumps to the roof tank showed readings increased to an abnormal extent. Checking the meter reading with the capacity of the duplex pump in operation at the time showed that, without allowing for slip, over 50 strokes



SUGGESTION TO PREVENT OBSTRUCTION IN METER

per minute would be necessary to supply the amount of water shown by the meter, while the pump was actually making only 30 strokes.

The chief decided there was an obstruction in the throat of the meter tube, and on examination, two rubber valves were found wedged there. After these were removed, the meter once more gave sensible readings and the operation of the pumps was also greatly improved. The diameter of the throat of the meter tube is 3¼ in. and that of the valves, about 4½ in.

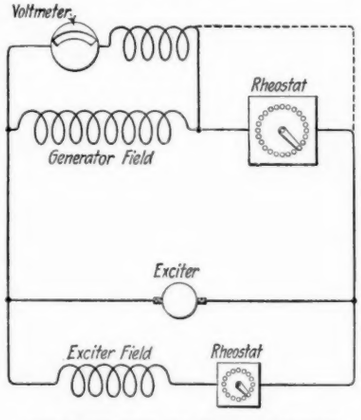
From this experience it seems to me that a strainer with a removable cover should be supplied by the manufacturers at a small additional cost. The mesh of the strainer need only be fine enough to stop anything too large to pass easily through the throat of the tube.

In the instance mentioned the valves would simply have dropped into the sediment chamber of the strainer and their presence would have caused no trouble.

Brooklyn, N. Y. Warren D. Lewis.

Exciter Voltmeter Varied

A difference of potential when applied to a series circuit that includes several devices of different resistances will distribute through the circuit accordingly as the resistance is distributed, the greater potential drop taking place across the greater resistance and the total drop being equal to the sum of the individual drops; provided, of course, that the circuit includes only ohmic resistances. If the total applied voltage be increased, all of the individual drops will increase, if the circuit includes no conductor that has a negative temperature coefficient and all devices have the same facilities for radiating the heat generated in them. Of course, if the heating due to the current is greater in some conductors than in others, or if there is greater difference in the temperature coefficients of the conductors involved, the resistances will not follow the same law of increase and the drops across them will not increase proportionately. An operator complained that the voltmeter used in connection with the exciter of an alternator varied in its indications and very erratically. The diagram represents the connections that were responsible for the irregular action of the voltmeter. The voltmeter, instead of being connected across the alternator field winding and field rheostat in series, as indicated by the dotted line, was connected across the alternator field winding only, as shown by the full lines. Under this condition the voltmeter could never indicate the exciter voltage except when the resistance of the alternator field rheostat was entirely cut out. Incident to regulation the field rheostat was, of course, subject to changes in response to the voltage requirements of load. Each change of the rheostat changed the relation of its resistance to the total resistance of the field circuit and to the resistance of the field winding. While the resistance of the field winding that the voltmeter spanned did not itself change, the change in the resistance of the rheostat would cause a redistribution of potential throughout the field circuit and would thereby affect the difference of potential across the field winding and voltmeter. Therefore every change of the alternator field rheostat produced a change in the voltmeter reading.



VOLTMETER IN SERIES WITH RHEOSTAT

The error of connection was excusable since of two wires that could be traced only with difficulty, one could very easily be handled instead of the other.

Schenectady, N. Y. E. C. Parham.

Inquiries of General Interest

A. S. M. E. Boiler Code—How can copies be obtained of the Boiler Code of the American Society of Mechanical Engineers? J. H. N.

The Code can be obtained in pamphlet form by addressing the secretary of the American Society of Mechanical Engineers, 29 West 39th St., New York City; price to members of the society, 40c., and to nonmembers, 80c.

Percentage of CO_2 Obtainable—What percentage of carbon dioxide (CO_2) may be expected under normal conditions of boiler-furnace practice, and what is the highest theoretical percentage? V. O. S.

The percentage of CO_2 obtainable in boiler-flue gases depends on the composition of the fuel. Under normal conditions of hand firing and average quality of coal it is possible to obtain 10 to 12 per cent. of CO_2 and theoretically 20 to 21 per cent.

Advantages of Low-Pressure Steam Heating—What are the advantages of a low-pressure over a high-pressure steam-heating system for a public building? J. S.

With a low-pressure system of steam heating the leading advantages over a high-pressure system are greater safety, lower temperature of heating surfaces, more positive regulation of the temperature of rooms whether by automatic regulation or by occupants of rooms, gravity return of the condensate to the boiler and the requirement of less skill and attention for operation than a high-pressure system.

Net Collapsing Pressure on Shell of Condenser—What is the collapsing pressure on the shell of a surface condenser carrying 26 in. vacuum? G. K.

The collapsing pressure would be the difference between the absolute pressure of the atmosphere and the absolute pressure within the condenser, which, in fact, would be the same as the pressure represented by a column of mercury 26 in. high. The pressure due to each inch of height of a mercury column is equivalent to 0.491 lb. per square inch, and therefore the net collapsing pressure would be $0.491 \times 26 = 12.766$ lb. per sq.in.

Depth of Holes in Hollow Stay-Bolts—To what depth should the telltale holes be drilled in hollow safety stay-bolts? W. I. N.

The purpose of the telltale hole will be served if it is drilled from the outside of the boiler beyond the least cross-sectional area of the stay-bolt. The best form of hollow stay-bolt is that having the hole throughout its full length, so it may be readily cleared by punching any obstruction out of the hole and to insure the benefit of the telltale hole in giving warning of rupture of the stay-bolt at any cross-section of its length.

Use of 220-Volt Motor on 110-Volt Circuit—How would the speed and power of a 220-volt motor be affected if used on a 110-volt circuit? W. P. M.

When used on a 110-volt circuit without rewinding, the speed of the 220-volt motor may be reduced as much as 25 per cent. and also the torque would be reduced. The reduction to half-voltage at the armature terminals would tend to decrease the speed to one-half, but as this to a certain extent would be offset by a lower voltage across the field, the weakened field current would tend to increase the speed, resulting in the reduction of the speed as stated.

Fire Cracks in Boiler Shells—What causes fire cracks in boiler shells and splitting at girth-seam rivets? J. R.

Fire cracks in boiler shells generally result from chilling the boiler plates by sudden admission of cold air through the fire-door and are more likely to occur when the plates become overheated because of the presence of oil or scale in the boiler, which prevents the rapid absorption of heat by the water. Cracks in the plates are also likely to occur from cold rolling in their manufacture or from too rapid bending of the plates in the forming rolls, and sometimes cracks develop at rivets as a result of injury to the plate received from the use of drift pins for fairing up the rivet holes.

Effect of Elevation on Draft of Chimney—What effect has elevation of a place on the force and volume of draft of a chimney of given size and height? C. A. W.

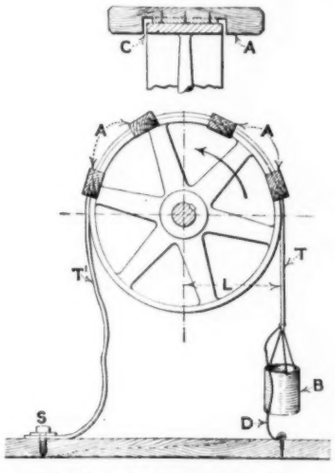
The force and volume of draft are due to the difference between the weights of the column of chimney gas and a column of the atmosphere of an equal height. For the same temperatures the ratios of their weights would be practically the same, but as their densities, or weights, per unit of volume are less at places of greater elevation above sea level, there would be less absolute difference of weight and consequently less force and volume of draft at the places of greater elevation.

Coking Method of Firing—How is boiler firing performed by what is known as the coking method? J. R.

In firing by the coking method, the fresh coal is placed just inside the furnace door, where it is allowed to remain until the heat from the fire drives the hydrocarbons out of the coal, and while this is in progress a considerable quantity of air is allowed to enter through the damper in the door. As the hydrocarbons are driven off, they become mixed with the air and are ignited in passing over the hotter portions of the fire. After the hydrocarbons are thus driven out of the coal, the remainder is in the form of coke that may be pushed back into the hotter portions of the fire where complete combustion is readily secured, after which a fresh supply of coal is placed in the front of the fire and the processes are repeated.

Strap Brake for Measuring Small Power—What is a simple form of brake for determining the power developed by a small motor? J. N. F.

A simple form of brake can be made by the employment of a piece of ordinary leather belting thrown over the driving pulley of the motor, as shown in the figure. A well-worn flexible belt, free to slip on the pulley face, will give the most uniform results. The belt should be narrower than the pulley face, and to provide against slipping sideways off the rim of the pulley, it should be tacked to three or four strips of wood AA notched so as to project over the pulley rim, as shown at C in the detail. Having rotation of the pulley in the direction indicated by the arrow, the side S of the belt is to be fastened to the floor, while from the other there is suspended a light-weight box B. A strong cord or wire D should be securely attached to the weighted side of the belt and connected with the floor to prevent the box from being pulled over the wheel, but this cord or wire must remain slack when the weights considered are in the box. Assuming that the belt tension at T due to the weight of the belt and empty box is equal to the tension at T′ due to the weight of the slack side of the belt, then when the motor is running, the weight in pounds, placed in the box and balanced, will be the net weight to be considered in computing the brake power wherein the distance L in feet, measured horizontally from the center of rotation of the pulley to the center of the belt, will be the lever arm, and the brake power developed may be determined by the usual formula



SIMPLE FORM OF BRAKE

$$\text{Brake hp.} = \frac{2\pi L\,NW}{33{,}000}$$

in which

$2\pi L$ = The distance in feet through which the end of the level arm would pass in one revolution;

N = Number of revolutions of the brake wheel per minute;

W = Weight in pounds balanced at the end of the lever arm L.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Report of Tests of a 30,000-Kw. Cross-Compound Turbine*

By H. G. Stott† and W. S. Finlay, Jr.‡

SYNOPSIS—Efficiency-tests report of the 30,000-kw. cross-compound steam turbines at the Seventy-Fourth Street Station[1] of the Interborough Rapid Transit Co., together with discussion of the paper.

The determining factors in the selection of power-generating equipment, either for an original installation or replacement purposes, present about as many variables as there are installations. Engineering experience has practically standardized the electrical end. The steam engine on the other hand, does not seem to have been capable of such standardization, as engineers are not agreed as to the most desirable superheat, pressure, etc., which must be considered both from the standpoint of reliability and boiler and engine economy. This thought is more than ever pertinent at this time, as it appears that we are on the eve of the employment of higher pressures and perhaps even new systems of power generation.

Further, the capacity of the machine to be selected and the relation of point of best steam consumption to maximum continuous load and provision for heavy overloads for limited periods are all dependent on load factors, diversity factors, etc., which present a new problem for every installation.

Without detailing at length the considerations which led to the selection of three 30,000-kw. units for the Seventy-Fourth Street Station of the Interborough Rapid Transit Co., a brief summation of the main questions involved will possibly be of interest and serve to classify the installation properly.

The prospective daily plant load, as was subsequently demonstrated, could be best provided for by units of 30,000-kw. maximum continuous capacity each. At that time there was assurance that machines of this capacity could be constructed, and furthermore, by the employment of the two-speed compound principle, as described more fully hereinafter, no hazards were involved, nor in fact would there be any departure from well-tried principles of construction. Ample steam supply was available by reason of development in the matter of relative boiler capacity coincident with the increase in efficiency of prime movers since the original construction of the plant in 1902. Economic considerations warranted the withdrawal of the horizontal-vertical double cross-compound engine-generator sets installed at that time, and their replacement by more efficient and larger units.[2]

The generally accepted essential requirements of a railroad plant were the next considerations—reliability, efficiency, cost.

In the development of turbine design, at the time the installation was being considered, possibly the simplest type of machine in many ways was the single-shaft, single-rotating-element turbine as a natural outgrowth of the generally accepted type developed in smaller capacities. Certain structural features, however, inherent with the larger capacity, tended to favor the division of the unit into two elements.

In a steam turbine, maximum centrifugal stresses are encountered at the exhaust end, where the greatest steam volume requires the greatest blade area. In the high-pressure blading, which uses steam of small specific volume, the best velocity ratio conducive to high economy cannot be met by the rotative speed as determined by the exhaust end. To avoid a compromise, the conditions are more readily satisfied by carrying out the expansion in two separate elements and avoiding congestion due to the high specific volume of the steam at 1 in. absolute exhaust pressure by the use of the lower speed, longer blades and the double-flow principle.

Reliability also seemed better served in the double-element machine by the shorter shaft and reduction of danger from temperature strains. Furthermore, there was the possibility, in an emergency, of operating either element of the unit alone, and that at a fairly high efficiency, the low-pressure element being available for service simply by the use of a by-pass. The use of the high-pressure end alone in the case of a bad breakdown on the low-pressure end was to be obtained by removing the low-pressure rotor and closing the shaft openings with special covers.

An extremely important consideration, favoring the divided unit in this particular case, was the matter of the weight of the parts to be handled by the crane which, in the case of the single unit, involved provision for additional crane capacity beyond that then available at the plant. To take care of such additional requirements would have necessitated the installation not only of a new crane, but reinforcements of the steelwork of a most elaborate nature and far in excess of that required for use in connection with the double unit.

In the matter of efficiency the double unit offered still more desirable possibilities, prominent among which was the relative flatness of its water-rate characteristic. Without entering into the matter of cost in this particular case, it may be said that the advantage was again with the double unit.

As a result of such considerations as the foregoing, the double-unit turbines were selected, together with condensing equipment designed to perform at a relatively high rate of efficiency, and including the necessary auxiliary apparatus, all of which is described in the following:

The high-pressure element is practically a typical single-cylinder reaction turbine containing 38 rows of blades. The low-pressure element is a turbine of the double-flow type. The casing is possibly unusual in its size and details of reinforcement. No blading is mounted directly in the casing,

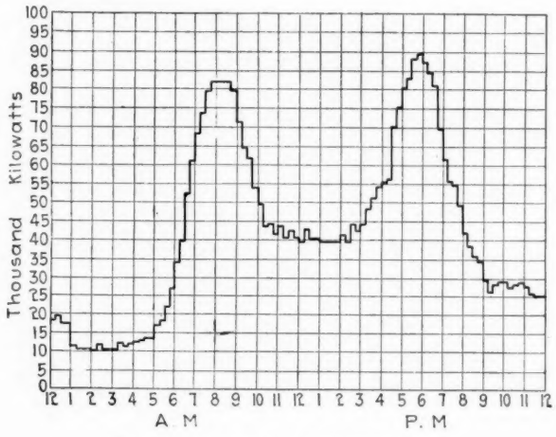


FIG. 1. TYPICAL LOAD CURVE

but through the intermediary of rings secured to it. The central portion of the low-pressure rotor consists of a drum to which are bolted two cast-steel shaft ends. These shaft ends have disks mounted upon them, carrying the last rows of blades.

Each turbine was erected upon a foundation of steel framework set in concrete, the steel structure alone being amply sufficient for support, the concrete being added to eliminate vibration.

Inasmuch as the high-pressure rotor was inherently of small diameter, it was possible to design it to operate at the maximum speed possible for 25-cycle service—that is, 1,500 r.p.m. The low-pressure turbine was designed for the next lower synchronous speed—namely, 750 r.p.m.—utilizing a four-pole generator.

The surface condensers which occupy most of the open space beneath the foundation framework consist each of two shells containing 25,000 sq.ft. of condensing surface apiece. The condenser shells are connected directly to the turbine outlets without an intermediate expansion joint. The weight of the condensers is carried by means of lugs, cast as a part of the shell, resting upon a number of spring jacks, which

*From a paper entitled "Report Upon Efficiency Tests of a 30,000-Kw. Cross-Compound Steam Turbine," read at the May, 1916, meeting of the New York Section of the American Society of Mechanical Engineers. For historical remarks by Mr. Stott introductory to the paper see page 733 of this issue.

†Superintendent of motive power, Interborough Rapid Transit Co., New York City.

‡Assistant to Mr. Stott.

[1]For a full page of data about the important equipment of this station—number of units, capacities, pressures, weights, performance guarantees, materials, etc.—see "Power," Apr. 25, 1915, p. 531.

[2]For a brief description of the important mechanical and economy features of these engines see "Power," Apr. 15, 1915, pp. 528 to 532.

are adjusted to carry the load without appreciable strain upon the turbine exhaust nozzles. Practically no restraint has been put upon longitudinal expansion of the low-pressure turbine by the circulating-water piping, which is fitted with rubber expansion joints. There is one expansion joint of copper in the steam-equalizing pipe between the shells.

The condenser auxiliaries were selected and installed with the prime factor of reliability continually in mind. The turbine-driven tri-rotor circulating pumps of 37,500 gal. per

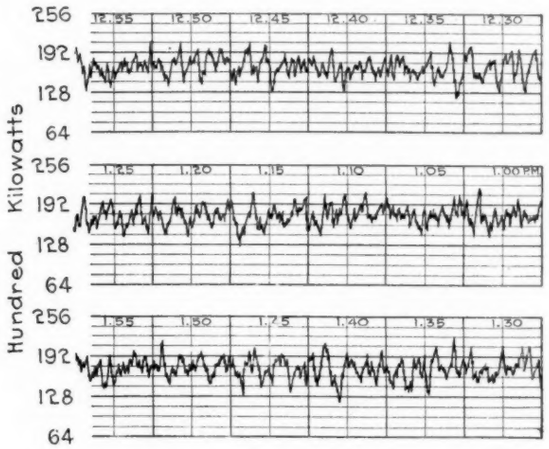


FIG. 2. PARTS OF WATT-HOUR METER CHARTS

min. capacity each were installed in duplicate, the reliability consideration being augmented by one of efficiency in the matter of the relative requirements for winter and summer use; during the winter months with cold water one pump is sufficient, but during the summer months high water temperatures necessitate two units in service.

The hotwell pumps were also installed in duplicate. A single rotary dry-vacuum pump was provided for each unit, with sufficient capacity to handle the dry air from two units, cross-connections being provided for the purpose.

The turbines are equipped with water-sealed glands. The gland water system has a small centrifugal turbine-driven pump for each unit, the piping for all of which is cross-connected through a common header. The water used for the turbine oil coolers returns the heat which is thus regained by way of the feed-water heaters.

The exhaust from the auxiliaries has been carried into the feed-water heaters which operate at low pressure. The shortage or excess of heat in this heater system, as the case may be, is compensated for or utilized by means of heat-balance valves operating between the auxiliary exhaust lines and the receivers between the turbine elements.

For test purposes one unit was selected as representative of the installation. Tests were made upon this unit by the use of standard methods, special provisions being taken to secure accuracy, some of which are detailed here:

High steam pressures were observed by the use of gages, in duplicate where of importance, such gages being calibrated before and after each test. Temperatures were observed by carefully calibrated thermometers, immersed in iron pipe wells, filled with mercury or oil, depending upon conditions. Wells were of ample depth, and correction was made for immersion. Low pressures were observed by the use of mercury manometers. Vacuum readings were made by the use of mercury columns provided with a vernier reading to $^1/_{100}$ in. The mercury in these columns was regularly cleaned and its specific gravity determined before and after test series and corrections made accordingly. Temperature corrections on mercury columns were provided for by the use of thermometers set in each column casing, and additional corrections were made for specific gravity, meniscus and barometer reading. The barometer used on the test was calibrated by reference to the U. S. Weather Bureau.

The condensate was weighed in tanks mounted upon two carefully calibrated platform scales. Platform scales were also used to measure drips and leakage. The unit was isolated so far as water and steam outlets were concerned. The large atmospheric relief valve necessarily remaining connected to the exhaust system was water-sealed, the seal being kept under continual observation by means of a window in the valve cover and interior illumination by electric light.

The unit output was obtained by means of three single-phase rotating standard watt-hour meters, one connected to each phase. These watt-hour meters were calibrated before and after the test series, which calibration included the current and potential transformers and showed no variation in excess of 0.2 per cent.

Each test was of three hours' duration with about a half-hour preliminary operation under test conditions. The load was controlled from the switchboard, the turbine operating upon the governor with hand throttle wide open. This subjected the turbine to the full swings of the railroad load, Fig. 1, as observed by means of a graphic wattmeter, which furnished an interesting index of such load variations. Table 1 gives a summary of the results of these charts, and Fig. 2 shows a typical section of a chart. At light loads these variations were more pronounced, approximating frequently 5,000 kw. either side of the average. A variation of 10,000 kw. total in half a minute was not uncommon, especially on loads of 16,000, 18,000 and 20,000 kw.

The test results have been shown graphically in Figs. 3, 4 and 5, upon bases of load and water-rate, load and thermal efficiency, and load and Rankine-cycle efficiency ratio. In conducting these tests, naturally every effort was made to maintain certain standard conditions under which guarantees had been given, and the test results as tabulated have been corrected to such standards. These standards represent what is probably the average of operating service. Corrections for variation from standard conditions were based upon curves as shown in Figs. 6, 7 and 8, which corrections were agreed upon prior to the commencement of tests.

In discussing and analyzing the test results, attention might be called to the following particular features:

1. With due allowance for scale of ordinates, the performance curves may be considered unusually flat, naturally conducive to high plant efficiency.

2. The dip in the curve between 22,000 kw. and 26,000 kw. is a peculiarity which was received at first rather skeptically, but which was later remarkably checked by repetition of tests throughout the range, including a special series under steady load made three months subsequent to the original series and given herein as Nos. 22-26 in the tables showing test summaries. Various theories have been advanced in connection with this dip and a series of special tests was made to investigate the relative action of the receiver between the two cylinders as a separator, and the velocities of the steam passing through it, with the idea that this might have some direct bearing upon this dip. Unfortunately, winter load de-

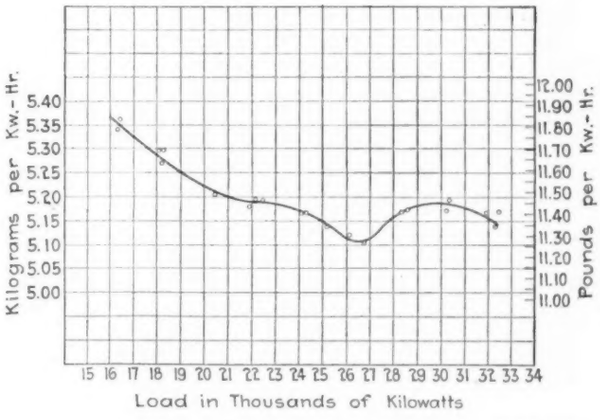


FIG. 3. WATER-RATE CURVE; GOVERNOR CONTROL

mands terminated further research work in this direction without any definite results having been obtained.

3. The turning up of the curve between 30,000 and 32,000 kw. is another peculiarity, accounted for by the turbine designer as follows:

> Concerning the supposed inconsistency in these tests—that is, the turning up of the efficiency curve between 30,000 and 32,000 kw.—consideration and figuring indicate that this is actually not an inconsistency, but a new experience. This turbine was designed for higher hydraulic efficiency than probably any machine heretofore built, thus approaching the crest of the efficiency curve. The overload capacity of the machine is small, or in other words, the amount the turbine is bypassed when the secondary valve opens is small, and the velocity ratio, therefore, is very little lower when full steam pressure is applied to the secondary inlet than when such pressure is applied to the primary inlet. Further, the hydraulic efficiency is nearly the same, so that the Rankine cycle at 32,000 kw. should not be more than 1 per cent. lower than at the point of best efficiency, namely 26,000 kw. The efficiency at the intermediate overload, say 30,000 kw., is somewhat worse than this, for while the blading and hydraulic efficiencies remain as high, there is a loss due to a certain portion of the steam expanding through the secondary valve to a lower pressure without doing work.

A number of tests were made upon the condensers and auxiliaries, separate and individual guarantees having been made upon the condenser proper, circulating pumps, dry vacuum pumps and hotwell pumps. Publication of the test data upon the pumps would simply show good efficient performance without unusual features. The results of the tests upon the condenser have been given, however, simply as a matter of information and interest in connection with the installation as a whole. The guarantees were based upon operation under maximum load, namely, about 32,000 kw., and the test was made with the turbine carrying as near that load as was possible. The duration of this test was three hours, the preliminary operation under test conditions, half an hour. The results have been shown in Table 3. No corrections were made in this test, as operating conditions approximated the guaranteed conditions very closely, with the exception that air leakage was high. The tests were made very shortly after the installation of the machine, and little opportunity had been given under operating requirements properly to eliminate this leakage, which has since been done. A permanent gasometer has recently been installed in connection with each unit, in order to observe air leakage at regular intervals and aid in its elimination.

TABLE 3. CONDENSER TEST, SUMMARY OF RESULTS

Pressure at throttle, abs....	220 lb. per sq.in.....	15.466 kg. per sq.cm.
Temperature at throttle....	487 deg. F.....	252.8 deg. C.
Superheat.....	97 deg. F.....	54 deg. C.
Load, average, kw.....	31,233.....	
Exhaust vacuum.....	28.61 in. hg.....	726.69 mm. hg.
Exhaust pressure, abs.....	1.39 in. hg.....	35.31 mm. hg.
Corresponding temp.....	89.4 deg. F.....	31.9 deg. C.
Mean temp. difference (log.)	12.9 deg. F.....	7.17 deg. C.
Heat transferred per hr....	316,000,000 b.t.u....	79,632,000 large calories
Heat transferred per unit surface per hr.....	6,330 b.t.u. per sq.ft.	17,150 large calories per sq.m.
Heat transferred per unit surface per hr. per deg. M.T.D.....	490 b.t.u. per sq.ft...	2,390 large calories per sq.m.
Air leakage per min.....	16.88 cu.ft.....	478 cu.dm.
Condensate per hr.....	357,060 lb.....	161,962 kg.
Temp. of hotwell water....	86 deg. F.....	30 deg. C.
Temp. of intake water.....	70.8 deg. F.....	21.5 deg. C.
Temp. of discharge water ..	80.9 deg. F.....	27.2 deg. C.
Rise.....	10.1 deg. F.....	5.7 deg. C.
Cir. water per min.....	64,700 gal.....	243,890 liters

Summarizing the results of these tests, it may be said that the performance in the cases of both turbine and condenser showed higher efficiencies than were guaranteed under contract, and the installation has proven to be thoroughly satisfactory in every particular, having fully realized the considerations which governed its selection.

Discussion of the Paper

Francis Hodgkinson[2]—The casual inspection of Figs. 3 and 5 would lead one to believe there are many inconsistencies. Both of these figures exhibit two rather curious reverse bends, one occurring between 22,000 and 26,000 kw. and the other between 27,000 and 32,000 kw. The latter lends itself to an explanation that has been quoted in the text of the paper. The reverse bend occurring between 22,000 and 26,000 kw. is not as easily explained. Its existence during the tests was doubted, and various errors were suspected. Repetition, however, of the tests made it certain that this vagary actually existed.

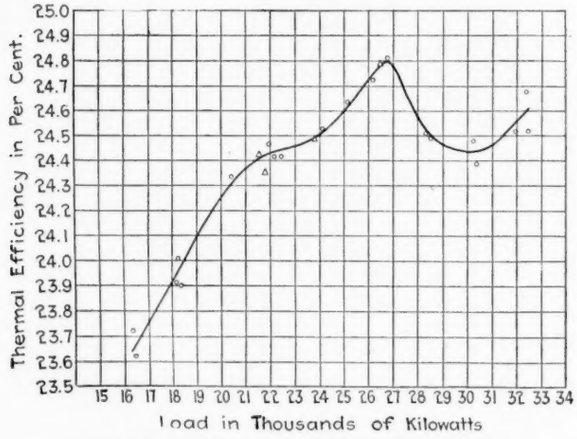


FIG. 4. THERMAL EFFICIENCY CURVE
O = Tests under governor control
Δ = Tests under throttle control

It is unfortunate that Fig. 3 did not have plotted thereon the total steam against kilowatts, because up to the point of best efficiency this should be a right line, and hence the performance of the machine may be best studied by using such a line.

Plotting this line through the averages of the test points, it is proved that the greatest deviation from the right line law amounts to 0.053 lb. per kw.-hr. in test No. 4.

It would look to be considerably more than this in Fig. 3, because of the large scale of the ordinates. The peculiarity, whether large or small, nevertheless exists, and it was supposed that the receiver separator installed between the high-pressure and low-pressure elements might offer an explanation.

It seemed in advance that there was an advantage in the cross-compounding scheme, in that there was an opportunity to effectively separate out the precipitated water between stages and relieve the low-pressure element of the friction of this inert fluid, and so an attempt was made to provide a satisfactory steam separator between the two turbine elements. Everyone was agreed that the most effective separator would be a vessel in which the steam velocities would be lowered to some 15 or 20 ft. per sec. Such a separator was, however, out of the question because of the enormous steam volumes involved. There is another, and what might be called opposite means, of separating moisture—that is, passing the steam at high velocity around curved surfaces and abstracting the moisture by centrifugal force. This was, in part at least, employed. The steam passes are so made that the water once separated will not again get into the current of steam flow.

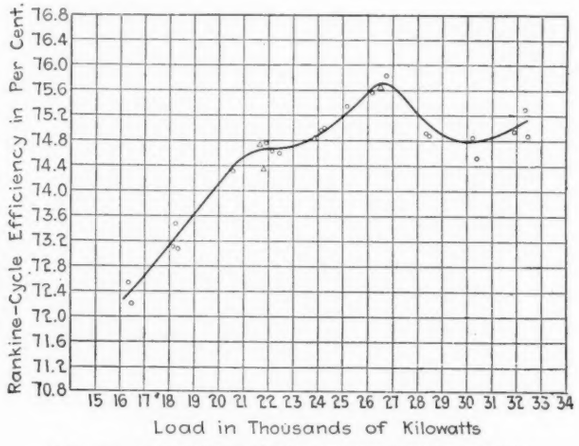


FIG. 5. RANKINE CYCLE EFFICIENCY RATIO
O = Tests under governor control
Δ = Tests under throttle control

During the tests calorimeter readings were made of the quality of the steam to the low-pressure element, and these indicated the separator to be effctive at minimum and maximum flows, with a point of ineffectiveness at moderate flow. It was proved, however, that these calorimeter readings were not to be relied upon, and in fact, any calorimeter readings are difficult to obtain under variable load conditions, because of the heat inertia of the calorimeter parts. Contemplation, however, of the separator makes it readily conceivable that with small flow, effective separation may be accomplished by virtue of the low velocity. It was ascertained that at maximum or heavy flow the separator was effective by reasons of centrifugal forces, and that there may be some intermediate and critical point where the water would sweep through without separating from the steam.

It is readily understood that it was not easy to provide space for a separator to effectively handle such volumes of steam.

At the conclusion of the test the builders of the turbine set about an investigation of their separator with a view of eliminating the peculiarity in the test results and generally betterng the performance. It is unfortunate that on account of the exigencies of power-house operation these could not have been brought to a conclusion.

These investigations were, of course, some interference, because the observations were useless except at quite steady loads.

It is further unfortunate that we only made observation of the pressure sweep through the separator and the quality of steam entering the low-pressure element and did not install means of increasing the quantity of water actually separated and discharged to the condenser well through the trap.

The moisture in percentage of pressure drop was plotted against the steam flow, the pressure drop being 2.8 in. between

[2]Engineer, turbine department, Westinghouse Electric and Manufacturing Co.

the high-pressure exhaust and the low-pressure inlet and 1.9 in. between the high-pressure exhaust and the bore of the separator, with 370,000 lb. per hr. flow.

It became evident that a gain in efficiency was to be obtained if the pressure drop were eliminated, even at the sacrifice of increased moisture. So the experiment was made of removing all the separator elements and substituting a switch plate in the riffles in the bore of the inlet to the separator. This gave a marked improvement in separation as well as in pressure drop. At the maximum flow the total pressure drop went from 2.8 in. to 1.1 in. and the moisture

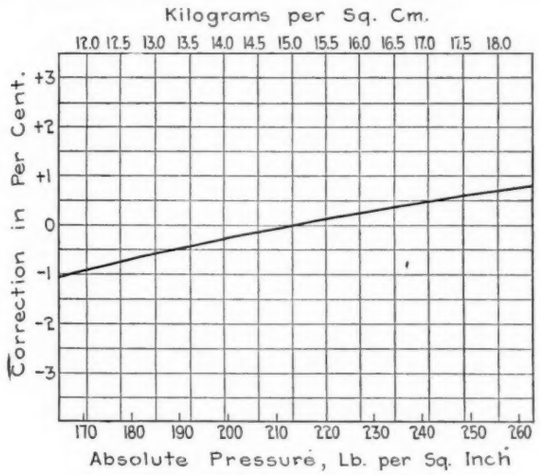


FIG. 6. PRESSURE CORRECTION CHART
Deviations from 215 lb. abs. and 120 deg. F. superheat

content from 3.55 per cent. to 2.41 per cent., so that further tests with the changed separator should show an improvement of some 1 to 1½ per cent.

A clause in the contract for these turbines provided that the test should be carried out with the load variation incidental to the elevated railroad and that, indeed, the load was anything but steady, as shown by Table 1, of the paper; and as shown, there was nearly 70 per cent. load variation, and there were single swings of 58 per cent., all of which, one would suppose, would affect the test results quite deleteriously; for while the heat absorbed by the turbine walls during an upward swing is all given back to the steam in a downward swing, the heat is taken from the steam which is at a higher pressure than when it is returned, which is, of course, a direct loss. However, tests with the pressure baffle did not show any particular difference.

R. J. S. Pigott[3]: Referring to the use of two speeds for high- and low-pressure elements, as noted in the paper, the inherent difficulty with the reaction-type turbine is that the high-pressure blading is always noticeably less efficient than the low-pressure blading. The hydraulic efficiency is probably not very different, but due to the shortness of the high-pressure blades, the leakage area over the tips is relatively much larger than for the low-pressure blades. The increase of speed in the high-pressure element increases the peripheral speed of any given size of drum; it is therefore possible, by still maintaining satisfactory blade speeds, to reduce the diameter of the drum and in this manner lengthen the blades for a given annular area; thus, of course, reducing the percentage of leakage area. The gain due to this feature alone is approximately 3 per cent. on the total water rate of the unit.

Referring to the use of the individual elements of the 30,000-kw. machine separately in emergency, in case the low-pressure element is used on bypass, it must be protected against overspeed in the same manner as the whole unit; therefore, the bypass was fitted with a 12-in. automatic valve operated from the overspeed governor of the turbine, in the same manner as the main throttle.

As to the foundations, it is to be noted that with turbine foundations of steel, when the structure has been designed for the allowable deflection, the question of strength is of little importance, since the foundation is then many times as strong as usual load-supporting structures. The maximum stresses, disregarding the reinforced-concrete envelope, amounted to about 7,000 lb. per sq.in.; the stiffening effect of the reinforced concrete is disregarded in figuring this deflection. Checking as closely as possible, the increase of strength due to the concrete envelope is about 40 to 45 per cent. additional.

[3]Mechanical engineer, Remington Arms Co.; formerly in charge of construction under Mr. Stott.

The main reason for employing a concrete coating was to destroy the excessive harmonic vibration occurring in plain steel structures. It has been found in those plants where uncovered steel foundations are in use (notably the Two-Hundred and First St. Station[4]), that with a turbine equally well balanced, harmonic vibrations are much more pronounced. The concrete envelope has an additional important function, namely, that of protecting the steel permanently against corrosion. The maximum deflection used in designing the steel structure was 0.020 in.

Referring to the condensers and their supports, the unusual feature of this installation is the large amount of movement between the two exhaust openings. In most installations at present, a single condenser has been employed, but as the change of length between the centers of the exhaust openings amounts in this 30,000-kw. turbine to 0.15 in. between the operating condition and the noncondensing condition, it would be practically impossible to use a single condenser without expansion joints between the turbine outlets and the condenser. In the original design, bronze single-bellows expansion joints were used on the piping, there being two 60-in., two 42-in., two 12-in. and a 36-in. connection. It was found, however, that the comparative stiffness of these joints caused the turbine casing to be distorted enough to endanger operation. When it is remembered that one of the 60-in. joints is at a distance of approximately 27 ft. from the center line of the turbine, it is clear that a tremendous leverage is exerted, and as far as the turbine casings are concerned the material cannot in any sense be considered rigid, and it responds to every stress with a corresponding distortion. Upon discovering this condition some of the joints were tested, and it was found that one of the 42-in. joints took 32,000-lb. load to compress it ⅛ in. It was evident that no metallic joint with a reasonable thickness of shell would be satisfactory. Rubber joints were therefore substituted, using the rubber in much the same manner as the bronze had formerly been employed, with the exception that cast-iron parts of the joints were so designed as to give the rubber maximum support against either pressure or vacuum, without interfering with its flexibility. As these rubber joints are only used in contact with water, there is no especial danger of oxidation and consequent short life. If any trouble from exterior oxidation should appear, it could no doubt be readily prevented by shellacking or paraffining the exterior of the joint.

Referring to the return of all waste heat possible, it is to be noted that the return of heat from the oil coolers amounts to nearly 1 per cent. of the total heat in the turbine, and the heat in the air from the turbine generators amounts to about 1 per cent. and is returned by carrying a discharge duct to the boiler-room cellar, where the warm air is used by the forced-

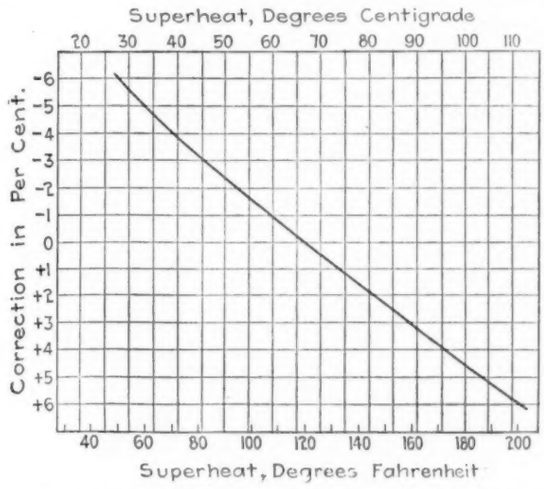


FIG. 7. SUPERHEAT CORRECTION CHART
Steam 215 lb. abs., superheat 120 deg. F., vacuum 29 in.

draft fans. The net gain (which is obtained practically without additional cost) is 1¾ per cent. of the total heat of the turbine. The question of steam- or electric-driven auxiliaries was settled by a graphic method of heat analysis covering all loads. In the Seventy-fourth St. station all auxiliaries are steam-driven. The total quantity of auxiliary steam increases slightly with the load on the main unit; it is therefore evident that at light load there will be excess exhaust steam over that required for the feed-water heater, and

[4]Main station of the United Electric Light and Power Co., New York City.

with heavy load there will be too little, the balance occurring at approximately 25,000 kw. on the main unit. The heat-balance valve compensates for this feature; at periods of light load it will be noticed that the receiver pressure is less than atmospheric; at such times the excess exhaust steam is free to be admitted into the low-pressure element and there produces a certain amount of power. At high loads the pressure in the receiver is higher than atmospheric, and the conditions are therefore right to deliver steam to the auxiliary exhaust systems, as noted. Considering the almost ideally automatic nature of the foregoing operations, the heat-balance valve designed by Mr. Hodgkinson is admirably simple. In the original layout no free atmospheric exhaust was to be provided; ordinary free exhaust being expensive, as the large-sized piping is used a few times during the life of the turbine. It was thought practicable to dispense with it by providing an additional automatic throttle valve controlled by a mercury tripper operated by the vacuum. This tripper was directly connected to the condenser, and it was in the form of an exaggerated mercury gage; if the vacuum fell below a predetermined amount, say 19 to 20 in., a float in the mercury was lifted, tripping a pilot valve which closed the main throttle. It was also so arranged that in starting up, the same method of safeguarding could be employed, operating at from 1 to 2 lb. above atmospheric pressure, the device automatically resetting itself for the vacuum conditions, as the vacuum rises in the condenser. An additional safeguard was in the form of a breakable diaphragm to the condenser discharge tunnel. However, a scheme was devised which obviated the necessity of this latter precaution.

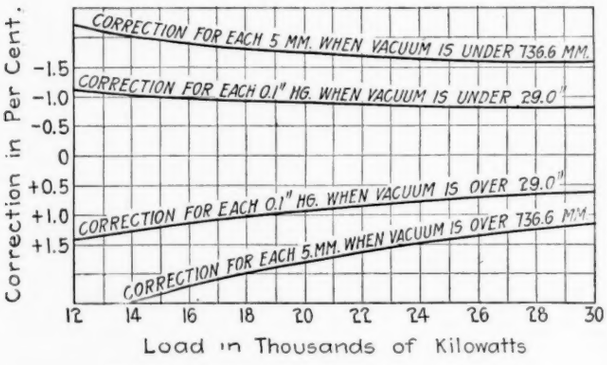


FIG. 8. VACUA CORRECTION CHART

Steam consumption corrections for each 0.1 in. hg. variation in vacuum from 29 in. hg. Per cent. of steam consumption at 29 in. hg. between limits of 28.5 in. hg. and 29.5 in. hg.

The auxiliary exhaust system is of such dimensions that it could be used as an atmospheric relief merely by enlarging a short section of the line from the turbine to the main exhaust. The old-engine 30-in. relief valves were remodeled, and the connection was made between the high- and the low-pressure cylinder. The loading of these valves, which of course have to stand pressure at the receiver of as much as 25 lb. abs., is accomplished partly with a hydraulic cylinder connected to the gland sealing water. In addition the dashpot on the other side of the valve was ring-packed and connected direct to the condenser, so that were the vacuum lost, part of the load would be automatically removed from the valve, assisting it to relieve the system promptly. As the discharge side of the valve was on the auxiliary exhaust system at 15½ lb. to 16 lb. abs., it was automatically steam-sealed; a water seal was therefore not necessary, but was installed as an additional precaution.

Referring to the variation of load during tests, as the Willans line up to best load is straight, and after this load is approximately straight at a different slope, the only point where a swinging load would have any effect would be at or near the best load, since with a straight Willans line the average water rate for swinging loads is the same as the water rate at a definitely fixed load equal to the average.

Referring to the investigations on the separator, the conditions found were not surprising, considering that practically similar experiences were obtained in the matter of moisture separation from low-pressure steam in connection with the low-pressure turbine at Fifty-ninth St. Station[5], in 1909 and 1910. The first separator installed on No. 1 unit at 59th St. was about 7x14 ft., full of baffles; the results were, practically no separation of moisture. The buffeting which the baffles received from the alternate exhausts of the engine soon loosened the rivets. When the baffles were removed, it was found that the quality immediately jumped from 92 per cent. to 96 per cent., using a plain tank.

Separators for Nos. 4 and 5 units were designed merely as plain tanks, 8 ft. in diameter and 32 ft. long, containing nothing but protection for the water in the bottom of the separator, and accomplished separation to 99 per cent. quality. From the consideration of the sizes of these separators and the amount of steam passed, it appears evident that similar separation is quite out of the question for the 30,000-kw. turbines. It seemed impossible with any baffle separator to remove anything but the water which is running along the sides or bottom of the pipe. The last 2 per cent. of "fog" can be removed by only one method—low velocity (not over 3,000 ft. per min.), and space, allowing time for the small drops to coalesce into larger drops and rain out by gravity.

PECULIARITIES OF WATER-RATE CURVE

Referring to the curve of water rates, Fig. 3, as Mr. Hodgkinson has remarked, some of the curious features of the curve are due to the exaggerated scale of the plotting. As he has also pointed out, it is desirable to make use of the Willans line for fairing the curve. When it is noticed that the total water consumption at any given load may vary 3,000 or 4,000 lb. between different tests, it seems apparent that the unusual curvature represented, up to 26,000 kw., is not justified, but the water-rate curve drawn from the straight Willans line will pass just as fairly through the points and give results more in accord with ordinary expectations. The same objection holds true in some degree to the remainder of the curve, although the first part of the rise from 27,000 to 31,000 kw. occurs nearly in accordance with the curve obtained direct from the Willans line. It is doubtful if there is such improvement in water rates in passing 31,000 kw. as the curve shows. Mr. Hodgkinson's point that the hydraulic efficiency of this machine is near the crest of the curve is perfectly true, but it must be remembered that when operating at maximum load on the secondary valve, there are only 32 stages in operation instead of 38, the first 8 being entirely blanketed at this time. Therefore, although the hydraulic efficiency of the stages may be as good as at best load, owing to the smaller number of stages on the same pressure range, it seems the efficiency could not be quite as good, especially as the first 8 stages are revolving idly in high-pressure steam, and the exit loss is increased.

Not only are Mr. Stott and his associates to be congratulated upon these remarkable results which have so closely justified the preliminary calculations, but especially Mr. Hodgkinson for the extraordinary excellence of design which, up to the present, represents the high-water mark of turbine engineering.

Norman G. Reinicker[6]: In the tabulation I notice that the vacuum decreases with the decreasing load. I was wondering if that was due to increasing air leakage, because in most turbines, as the load decreases—that is, as the quantity of steam to be condensed decreases—the condenser would give better vacuum, because it is running under load.

W. S. Finlay, Jr.: That is explained in the paper.

N. G. Reinicker: The paper says that this leakage has been corrected, more or less, and I was wondering what it is at these lower loads; how fast it increases. I was also wondering what the station load factor is and what the economy of the station was with the turbines all running and using the exhaust steam into the main turbine and the heat from the generator going back into the boilers.

The Chairman: It would be better for the authors of the paper to answer these queries in their closing.

Seley Haar: I should like to have Mr. Stott or Mr. Finlay in summing up state whether any attempt was made to use the pitot-tube type of steam heater, as a check on their methods of measuring the water in the feed.

W. S. Finlay, Jr.: We used the venturi type and not the pitot, and they checked up very closely—about ½ per cent. was the actual variation.

John P. Sparrow[7]: In line with Mr. Stott's historical review, I will give a few items in connection with the plant of the New York Edison Co.

Of the two Waterside stations, one was built in 1900 and one in 1906. The original equipment in those stations was 152,650 hp. of boilers, and they were designed for hand-firing, the grates burning No. 3 buckwheat coal. The development that has taken place in turbines and in the high capacities has resulted in an increase in capacity over the original 140,000 kw. to 250,000 kw., and we still have capacity for 100,000 kw. more, only going to the moderate rating of 250 per cent., which means we have between 450,000 and 500,000

[5]Station of the Interborough Rapid Transit Co., New York City, supplying power for the subway system.

[6]Assistant chief engineer, New York Edison Co., formerly with the Detroit Edison Co.

[7]Chief engineer, New York Edison Co

kw. available boiler capacity in those boilers originally installed.

The coal consumption of 3 or 3.5 lb. of coal per kilowatt-hour in the old No. 3 buckwheat days, was good. Today if we get down to 1.8 lb. of coal per kilowatt-hour, that is only a fair performance, and Mr. Stott, with his equipment and with the present efficiencies, can, I have no doubt, quote a better figure.

I would like to ask Mr. Stott, briefly emphasizing what Mr. Reinicker asked, as to the thermal efficiency of the station. We are interested in large turbines, we are all interested in high boiler capacity, and we have quoted results of 16, 17, 17¼, and in some case 17.5 per cent. thermal efficiency. As I understand the Seventy-fourth St. Station, these three

find that with one of the new boilers we can deliver as much steam to the plant as we can with any three of the old boilers.

Francis Hodgkinson: I would like to reply to Mr. Pigott as to the matter of the hydraulic efficiencies of blades. Perhaps that term may be a misnomer. I mean the efficiency of the turbine elements which result in a study of the velocity and blade angles, etc. With the velocity of the blade divided by the velocity of steam and, theoretically, with ordinary angles such as are usually employed, we reach the maximum with about 90 per cent. velocity-ratio.

All turbines are designed for a point of best steam consumption somewhere, regardless of what their rating may be. Their rating nowadays means little, and properly so. There

TABLE 1. SWINGING LOADS DURING TESTS AS RECORDED BY GRAPHIC CHART

		Average	From Graphic Load Chart									
Test No.	Duration of Test, Hr.	Kw. Load by Watt-Hour Meter A	Minimum Load During Test B	Maximum Load During Test C	Difference in Kw. of Max. and Min. Loads C—B = D	Per Cent. of Average Load D / A	Average of Difference Between Minimum and Maximum Load During Each 2½ Min. Period During Test E	Per Cent. of Average Load E / A	Max. Single Load Swing During Test F	Per Cent. of Average Load F / A	Length of Swing F in Sec.	Time of Test
1	3	16,342	10,100	20,800	10,700	65.5	6,080	37.2	8,700	53.3	21	12.00-3.00
2	3	16,447	9,800	20,300	10,500	63.8	6,200	37.7	8,200	49.8	40	12.00-3.00
3	3	18,158	12,800	23,000	10,200	56.2	5,680	31.3	7,800	43.0	34	1.00-4.00
4	3	18,273	10,400	23,100	12,700	69.5	5,860	58.6	10,600	58.0	41	12.00-3.00
5	3	18,317	11,600	22,000	10,400	56.8	6,360	34.7	9,400	51.3	24	12.00- 3.00
6	3	20,472	15,600	22,600	7,000	34.2	3,810	18.6	6,500	31.8	42	2.00- 5.00
7	3	21,925	14,900	26,100	11,200	51.1	3,740	17.1	6,650	30.3	26	8.00-11.00
8	3	22,150	16,500	27,400	10,900	49.2	4,200	19.0	6,500	29.4	24	8.00-11.00
9	3	22,432	16,900	25,400	8,500	37.9	4,580	20.4	7,400	33.0	33	8.00-11.00
10	3	24,137	18,700	27,800	9,100	37.7	3,700	15.3	6,900	28.6	36	8.00-11.00
11	3	24,200	20,500	27,600	7,100	29.3	3,160	13.1	4,300	17.8	15	8.00-11.00
12	3	25,170	19,700	27,400	7,700	30.6	3,260	13.0	5,500	21.9	12	10.30- 1.30
13	3	26,153	20,200	28,900	8,700	33.3	3,080	11.8	5,400	20.6	12	8.00-11.00
15	3	28,378	21,000	32,700	11,700	41.2	3,700	13.0	5,400	19.0	1	3.30- 6.30
16	3	28,517	23,200	29,700	6,500	22.8	2,200	7.7	4,000	14.0	11	3.30- 6.30
17	3	30,232	24,400	32,000	7,600	25.1	3,040	10.1	4,500	14.9	12	3.30- 6.30
18	3	30,397	24,600	32,800	8,200	27.0	2,780	9.1	5,600	18.4	2	3.30- 6.30
19	3	31,986	29,400	35,200	5,800	18.1	2,450	7.7	3,700	11.6	9	4.00- 7.00
20	3	32,348	32,900	28,300	4,600	14.2	2,490	7.7	3,100	9.6	5	4.00- 7.00

Test 19-22 inclusive—no chart taken.

TABLE 2. SUMMARY OF RESULTS, TURBINE TESTS

Test No.	Abs. Steam Pressure at Throttle Lb. per Sq. In.	Abs. Steam Pressure at Throttle Kg. per Sq. Cm.	Steam Temperature at Throttle Deg. Fahr.	Steam Temperature at Throttle Deg. Cent.	Superheat Deg. F.	Superheat Deg. C.	Abs. Steam Pressure Primary Inlet Lb. per Sq. In.	Abs. Steam Pressure Primary Inlet Kg. per Sq. Cm.	Abs. Steam Pressure Exhaust Hp. Lb. per Sq. In.	Abs. Steam Pressure Exhaust Hp. Kg. per Sq. Cm.	Abs. Steam Pressure Inlet L.P. Lb. per Sq. In.	Abs. Steam Pressure Inlet L.P. Kg. per Sq. Cm.	Exhaust Vacuum In. Hg.	Exhaust Vacuum Mm. Hg.	Exh. Pres. Referred to 30-In. Bar, 762 Mm. 58.1 Deg. Fahr., 14.5 Deg. Cent. In. Hg.	Exh. Pres. Referred to 30-In. Bar, 762 Mm. 58.1 Deg. Fahr., 14.5 Deg. Cent. Mm. Hg.	Load Average, Kw.	Water per Hr. Lb.	Water per Hr. Kg.	Water* Rate Lb. per Kw.-Hr.	Water* Rate Kg. per Kw.-Hr.	Rankine* Cycle Efficiency Per Cent.	Thermal* Efficiency Per Cent.
1	224.3	15.78	493.2	256.2	101.6	56.5	148	10.41	10.6	0.745	10.2	0.717	28.411	721.63	1.589	40.37	16,342	192,350	87,250	11.770	5.339	72.540	23.72
2	223.3	15.7	486.8	252.7	95.8	53.2	144	10.12	11.8	0.83	10.0	0.703	28.52	724.41	1.48	37.59	16,447	194,431	88,194	11.822	5.362	72.222	23.62
3	224.1	15.77	485.7	252.1	94.2	52.3	166	11.68	....	.....	11.0	0.773	28.539	724.9	1.461	37.1	18,158	212,029	96,176	11.677	5.297	73.118	23.91
4	223.3	15.7	489.6	254.2	98.4	54.7	159	11.18	11.8	0.83	11.5	0.809	28.471	723.15	1.529	38.85	18,273	212,330	96,313	11.620	5.271	73.476	24.02
5	222.6	15.65	496.5	258.05	105.7	58.7	163	11.46	11.8	0.83	10.8	0.759	28.588	726.15	1.412	35.85	18,317	213,976	97,059	11.682	5.299	73.086	23.90
6	220.4	15.5	510.9	266.1	121.0	67.3	175	12.31	12.7	0.893	12.0	0.844	28.586	726.08	1.414	35.92	20,472	234,930	106,564	11.476	5.206	74.400	24.33
7	219.8	15.46	519.3	270.7	129.5	72.0	182	12.8	12.8	0.9	12.5	0.879	28.683	728.55	1.317	33.45	21,925	250,407	113,585	11.421	5.181	74.757	24.46
8	221.8	15.6	518.8	270.5	128.0	71.1	185	13.01	13.5	0.949	12.8	0.9	28.791	731.3	1.209	30.7	22,150	253,388	114,937	11.439	5.189	74.630	24.41
9	221.1	15.54	501.5	260.8	111.2	61.8	187	13.15	....	.....	13.0	0.914	28.744	730.1	1.256	31.9	22,432	256,716	116,446	11.444	5.191	74.606	24.41
10	221.3	15.56	513.9	267.7	123.5	68.6	194	13.64	14.2	0.998	14.1	0.992	28.902	734.1	1.098	27.9	24,137	274,971	124,727	11.392	5.167	74.947	24.52
11	220.3	15.49	512.9	267.2	123.0	68.4	196	13.78	13.7	0.963	13.6	0.956	28.882	733.6	1.118	28.4	24,200	275,593	125,009	11.388	5.166	74.973	24.52
12	220.3	15.49	496.8	258.2	106.8	59.4	197	13.85	14.6	1.027	14.1	0.992	28.896	734.0	1.104	28.0	25,170	285,184	129,359	11.330	5.139	75.357	24.63
13	217.9	15.32	524.3	273.5	135.2	75.2	211	14.84	14.5	1.019	14.7	1.033	28.761	730.53	1.239	31.47	26,153	295,373	133,981	11.294	5.123	75.597	24.72
14	224.0	15.76	500.0	260.0	108.5	60.3	215	15.12	....	.....	15.0	1.055	28.862	733.1	1.138	28.9	26,740	301,035	136,549	11.258	5.107	75.840	24.81
15	221.6	15.58	510.0	265.5	119.5	66.4	210	14.77	16.5	1.16	15.9	1.118	28.862	733.1	1.138	28.9	28,378	323,353	146,673	11.395	5.169	74.928	24.51
16	220.0	15.47	515.5	268.6	125.6	69.8	211	14.84	16.6	1.167	16.2	1.138	28.812	731.82	1.188	31.18	28,517	325,072	147,453	11.399	5.171	74.901	24.49
17	220.3	15.49	513.0	267.2	123.0	68.4	210	14.77	16.5	1.16	16.8	1.181	28.78	731.01	1.22	30.99	30,232	344,787	156,395	11.405	5.173	74.862	24.48
18	217.2	15.28	518.0	270.0	129.1	71.8	206	14.48	17.3	1.216	17.2	1.208	28.801	731.55	1.199	30.40	30,397	348,018	157,861	11.449	5.193	74.574	24.39
19	217.1	15.27	515.4	268.6	126.6	70.3	212	14.9	18.5	1.301	18.5	1.301	28.623	727.03	1.377	34.97	31,986	364,280	165,237	11.389	5.166	74.968	24.52
20	218.2	15.33	519.5	270.8	130.4	72.5	209	14.7	17.6	1.237	18.2	1.28	28.75	730.25	1.25	31.75	32,348	366,572	166,277	11.332	5.140	75.343	24.64
21	216.3	15.21	513.3	267.4	124.9	69.4	209	14.7	18.1	1.272	18.9	1.328	28.765	730.63	1.235	31.37	32,490	370,335	167,984	11.398	5.1701	74.908	24.52
22	222.8	15.66	518.0	270.2	127.0	70.5	182	12.78	13.9	0.976	13.0	0.913	28.55	724.70	1.45	36.83	21,610	246,661	111,850	11.414	5.179	74.733	24.42
23	224.0	15.76	508.0	264.5	116.5	64.5	186	13.05	14.2	0.998	13.3	0.934	28.53	724.67	1.47	37.23	21,800	250,134	113,600	11.474	5.207	74.412	24.35
24	222.9	15.67	512.0	266.7	121.0	67.3	194	13.66	14.8	1.042	13.9	0.976	28.92	734.57	1.08	27.43	23,890	272,335	123,600	11.400	5.170	74.824	24.48
25	223.1	15.67	515.4	268.9	124.3	69.0	213	14.98	16.2	1.139	15.1	1.062	28.85	732.8	1.15	33.15	26,505	298,810	135,600	11.274	5.114	75.660	24.79

* As corrected to 215 lb. per sq.in., or 15.1 kg. per sq.cm. abs. press.; 120 deg. F. (66.6 deg. C.), superheat; 29 in. hg. (736.6 mm.)

30,000-kw. units carry practically the entire output of the station therefore, any figures which Mr. Stott might give would lead up to the best practice to date.

J. W. Thomas[8]: With relation to the history given by Mr. Stott, he brought out that the larger capacity that he has been able to get on the boiler equipment is due not so much to the increased efficiency of the turbine as it is to the increased capacity of the boiler. Today, with the underfeed stoker, it is possible, and I believe Mr. Stott will agree with me, to get better than 300 per cent. of the rated capacity in everyday operation.

We have just installed at New Orleans a 900-hp. Babcock & Wilcox boiler, underfeed stoker, on which in everyday operation with the ordinary operating crew we are able to get 450 per cent. of the rating.

The boiler is a duplicate of twelve others in the plant, put in from 1904 to 1908, equipped with overfeed stokers. We

[8]Engineer, United Gas and Electric Engineering Corporation, New York.

are different kinds of power plants; the variable street-railway loads, for instance, where you want a machine of 1,000-kw. capacity and may want to get away with 2,000 kw. on a peak. If a turbine is designed for its best point of efficiency at 1,000 kw. and it is to be used in a big station where practically it carries nearly the maximum load all the time, we set the point of best steam consumption a little below that. You might design a turbine for carrying a given load and have a moderate overload capacity. You can put the hydraulic efficiency wherever you please. Let us say under the normal load, before you resort to bypassing, you select a point on this efficiency curve and then cut out a portion of the high pressure turbine and bypass it; there will be fewer elements in series. The velocity-ratios go down, and the hydraulic efficiency remains the same.

Henry G. Stott: There are a few questions I would like to answer. Mr. Roberts asked what would happen if we added 25 per cent. overload on the generator. Obviously, the turbine would begin to shut down, nothing else would happen, there would be a reduction in speed. The actual speed limit, the

limit of load it will carry, without loss of speed, is apparently about 33,000 or 34,000 kw. We have actually carried 32,000 on the test and maintained full standard frequency, 25 cycles; but beyond that, somewhere between 33,000 and 34,000, the turbine begins to drop in speed. If you added 25 per cent. to the 30,000-kw. load, you would bring the speed down between 15 and 20 per cent. This, of course, will happen with any of the large machines designed today.

In regard to the coal per kilowatt-hour on the engines before the turbines were put in, we were running about 2.5 lb. of coal for the plant, averaged at the end of each month; that includes everything—all auxiliaries, coal-handling apparatus, etc., etc. In other words, the total coal used per month divided by the kilowatt-hours output, or delivered to the feeders leaving the power house, what we call the net output, would result in a figure of approximately 2.5 lb. of coal per kilowatt-hour.

Since the turbines were put in, and while they have been carrying practically all of the load too, with the steam mains leading to the engines still alive for emergency purposes, the coal consumption is approximately 1.5 lb., a drop of 1 lb. per kw.-hr. The thermal efficiency of the station for the monthly average is approximately 17 per cent.

In regard to the question of reheaters between the high- and low-pressure turbines, that practice has been abandoned, as previous investigations in connection with the engines show no improvement, and there probably is none.

The question was asked as to why the vacuum went down with the load, and that is undoubtedly due to air which at that time we were unable to overcome, but later, by the use of a gasometer we have been able to eliminate that trouble.

The thanks of the meeting was extended to Messrs. Stott, Finlay and Hodgkinson.

Massachusetts Board of Boiler Rules Hearing

The Massachusetts Board of Boiler Rules held a hearing at the State House, Boston, on Friday, Apr. 28, to consider two amendments to the rules as formulated by the board. The first of these related to fusible plugs. Amendment 4 of the present rules now reads:

Fusible plugs as required by Section 20, Chapter 465, Act of 1907, shall conform to the following specifications: (See diagram on p. 55.)

All fusible plugs shall consist of a bronze casing, holes in which shall be reamed and tinned before being filled, and shall be filled with 99 per cent. pure tin.

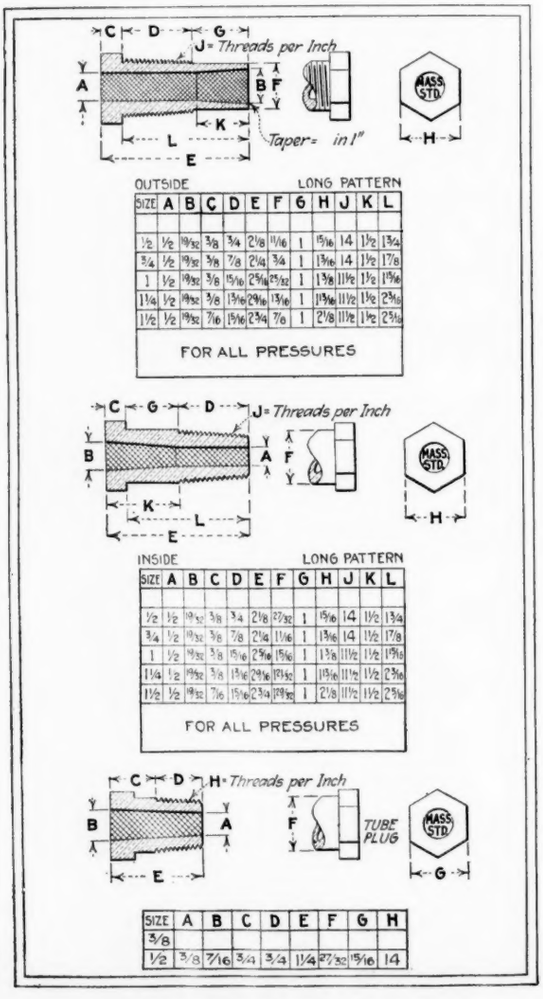
All fusible plugs shall be stamped by the manufacturers with their names across the face of the plugs, with letters not less than ⅛ in. in height and letters Mass. Std.

The outside diameter of the plug is to be of the standard pipe threads.

No fusible plug shall be used for a longer period than one year.

It is proposed to amend the second paragraph of this amendment to read:

All fusible plugs shall consist of a bronze casting, holes in which shall be reamed and tinned before being filled, shall have an even taper from end to end not less than ⅝

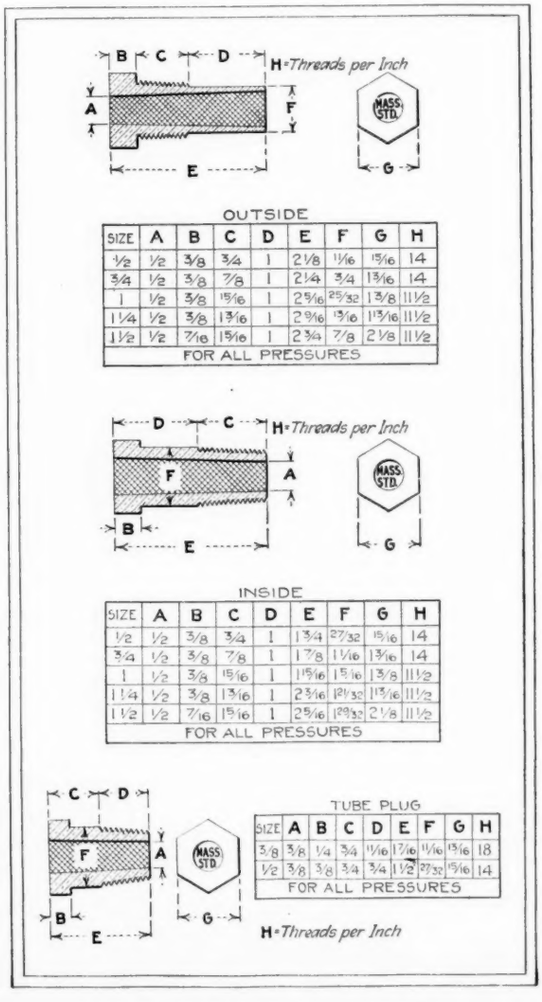


SIZE	A	B	C	D	E	F	G	H	J	K	L
½	½	19/32	3/8	¾	2⅛	11/16	1	15/16	14	1½	1¾
¾	½	19/32	3/8	⅞	2¼	11/16	1	1 3/16	14	1½	1⅞
1	½	19/32	3/8	15/16	2 5/16	25/32	1	1 3/8	11½	1½	1 15/16
1¼	½	19/32	3/8	1 3/16	2 9/16	1 3/16	1	1 3/16	11½	1½	2 3/16
1½	½	19/32	7/16	15/16	2¾	⅞	1	2⅛	11½	1½	2 5/16

SIZE	A	B	C	D	E	F	G	H	J	K	L
½	½	19/32	3/8	¾	2⅛	27/32	1	15/16	14	1½	1¾
¾	½	19/32	3/8	⅞	2¼	11/16	1	1 3/16	14	1½	1⅞
1	½	19/32	3/8	15/16	2 5/16	15/16	1	1 3/8	11½	1½	1 15/16
1¼	½	19/32	3/8	1 3/16	2 9/16	1 21/32	1	1 5/16	11½	1½	2 3/16
1½	½	19/32	7/16	15/16	2¾	1 29/32	1	2⅛	11½	1½	2 5/16

SIZE	A	B	C	D	E	F	G	H
3/8								
½	3/8	7/16	¾	¾	1¼	27/32	15/16	14

FIG. 1. THE FORMER TABLE



SIZE	A	B	C	D	E	F	G	H
½	½	⅜	¾	1	2⅛	1⅛	15/16	14
¾	½	⅜	⅞	1	2¼	¾	1 3/16	14
1	½	⅜	15/16	1	2 5/16	25/32	1⅜	11½
1¼	½	⅜	1 3/16	1	2 9/16	1 3/16	1 13/16	11½
1½	½	7/16	1 5/16	1	2¾	⅞	2⅛	11½

SIZE	A	B	C	D	E	F	G	H
½	½	⅜	¾	1	1¾	27/32	15/16	14
¾	½	⅜	⅞	1	1⅞	1 1/16	1 3/16	14
1	½	⅜	15/16	1	1 15/16	1 5/16	1⅜	11½
1¼	½	⅜	1 3/16	1	2 3/16	1 21/32	1 13/16	11½
1½	½	7/16	1 5/16	1	2 5/16	1 29/32	2⅛	11½

SIZE	A	B	C	D	E	F	G	H
⅜	⅜	¼	¾	1/16	1/16	1/16	1/16	18
½	⅜	⅜	¾	¾	1½	27/32	15/16	14

FIG. 2. THE TABLE AS AMENDED

in. to the foot, and shall be filled with 99 per cent. pure tin,
and to substitute for the diagram in the present copy of the rules, which is shown in Fig. 1, that shown in Fig. 2.

James Stewart, of Worcester, pointed out that under the law as it stood and under the proposed amendment plugs must conform to dimensions given in the specifications in all particulars. This precluded the use of plugs which were more generous in some of their dimensions and therefore even better than those called for by the rules. No court would construe as an offense the use of these better plugs, but it would be well to so frame the amendment that it would have some elasticity in the right direction.

The other amendment related to par. 28 of Section 4, Part 3, on page 38 of the 1916 edition of the Rules, which paragraph deals with the segments of heads of horizontal return-tubular boilers necessary to be braced. It is proposed to add to the section:

When an unstayed flat surface greater than 5 inches (5 in.) is desired, measured from the edge of tube to the outside of flange on tube sheet, the following formula shall be used:

$$p = \sqrt{\frac{200t^2}{P}}$$

p = Distance, in inches, from the outside edge of the tube to the outside edge of the flange on tube sheet, in a straight line;
t = Thickness of plate, in sixteenths of an inch;
P = Maximum allowable working pressure per square inch;
200 = Constant.

The 5 in. designated is intended to cover the 3 in. allowed for the flange and 2 in. for the supporting power of the tubes, but S. F. Jeter, chief engineer of the Hartford Steam Boiler Inspection and Insurance Co., pointed out that the phraseology of the amendment did not entirely convey this idea and was asked to submit a brief with his suggestions.

ENGINEERING AFFAIRS

The American Society of Safety Engineers will hold its regular monthly meeting Wednesday, May 31, 1916, at 8 p.m., in the Engineering Societies' Building, 29 West 39th St., New York City.

Railway Electrification—A. L. Rohrer, electrical superintendent of the Schenectady Locomotive Works, gave a short history of railway electrification on Apr. 8 at a meeting of the Minneapolis and St. Paul Engineering societies, at the University of Minnesota. George H. Hill, of the railway and traction department of the General Electric Co., declared that economy of operation and greater efficiency had been attained by electrification of the Chicago, Milwaukee & St. Paul Ry. through the Belt, Rocky and Bitter Root Mountains.

University of Wisconsin Summer Session—The eighteenth annual summer session of the College of Engineering of the University of Wisconsin will be held at Madison during the six weeks beginning June 26. Special courses will be given in chemistry; electrical, steam and hydraulic engineering; gas engines; machine design; mechanical drawing; mechanics; shopwork and surveying. All courses given in the university summer session are open to engineering students. Special courses have been arranged for engineering, manual arts and vocational teachers. For information address F. E. Turneaure, Dean, Madison, Wis.

University of Illinois Receives Steam Specialties Exhibit—The Department of Mechanical Engineering of the University of Illinois has received from the Crane Co., Chicago, an unusually complete and attractive exhibit of the product of this company. It includes a number of large valves, steam traps and other power specialties mounted on polished cast-iron floor pedestals, together with an oak display case mounting samples of the smaller valves, fittings and specialties. The completeness of the exhibit and the attractive way in which it has been prepared to permit the operation of the devices to be demonstrated, make the collection of unusual value to students in mechanical engineering.

Study of Stresses in Boiler Heads—Prof. G. A. Goodenough, of the Department of Mechanical Engineering of the University of Illinois, has undertaken an analytical study of the stresses in convex boiler heads, and it is expected that actual experimental work to verify the analytical results will be undertaken by Prof. H. F. Moore, of the Department of Theoretical and Applied Mechanics. It is hoped that this investigation may lead to a more rational and safer design of boiler heads. The Babcock & Wilcox Co. has presented to the department a 42-in. cylindrical drum to be used in the investigation. One head has a manhole, and the other—the head to be tested—is fastened to the shell by specially designed tapered bolts to facilitate its removal and replacement by heads of different design. The shell of the drum is unusually heavy to permit pressures to be carried above those normally used in practice.

OBITUARY

E. L. CORTHELL

Dr. Elmer L. Corthell, president of the American Society of Civil Engineers, and one of the most noted of his profession in this country, died in Albany, N. Y., May 16, in his seventy-seventh year.

WILLIAM STANLEY

William Stanley, a prominent electrical engineer and inventor, died on May 14 at his home in Great Barrington, Mass. He was born in Brooklyn, N. Y., Nov. 22, 1858. He was graduated from the Williston Seminary at Easthampton, Mass., and spent a short period in Yale University with the class of 1881. Becoming interested in electricity, he left the university to take up experimental work, and in this science he was self-educated.

Mr. Stanley was the inventor of a great many electrical devices that have been very conspicuous in the development of the electrical industry. Chief among his inventions were those which made possible the long-distance transmission of electrical energy. For his attainment in electrical engineering he was awarded the Edison Medal in 1912, which is the highest award to an electrical engineer in this country.

During the period 1885-8 Mr. Stanley was chief engineer of the Westinghouse Electric Co.; in 1890 he became the head of the Stanley Electric Manufacturing Co., remaining there for five years. Then he was a consulting engineer for three years, in 1898 becoming the head of the Stanley Electric Instrument Co., where he remained until 1903. He was a member of the Institute of Electrical Engineers, London, England, and the American Institute of Electrical Engineers.

PERSONALS

George Mesta, president of the Mesta Machine Co., Pittsburgh, Penn., has been elected first vice-president of the National Metal Trades Association.

Forrest W. Manker has accepted a position with the B. F. Sturtevant Co., Cleveland, Ohio. He was until recently connected with the Moulton Engineering Corp., of Portland, Me.

Walter J. Bitterlich has been appointed plant engineer of the Hood Rubber Co., Watertown, Mass. He was formerly connected with this company in the capacity of assistant master mechanic.

P. A. Beaman has been appointed assistant superintendent of George W. Prentiss & Co., Holyoke, Mass. He was formerly with the Manufacturers Equipment Co., Boston, Mass., as mechanical engineer.

L. Guy Long, until recently affiliated with the Fairbanks-Morse Electric Manufacturing Co., Indianapolis, Ind., as mechanical engineer, has accepted a position with the Gibson Co., of the same city.

F. L. Gilman, assistant general superintendent of the Western Electric Co., Chicago, Ill., has been appointed one of the assistant chief engineers of the company, with headquarters in New York.

Burton C. Fonda, until recently in the employ of the Thomas Crimmins Contracting Co., New York, has become associated with the General Electric Co., Schenectady, N. Y., in the turbine department.

W. Clement Brooke, until recently inspector of the East Orange (N. J.), Water Department, has accepted the position of assistant construction engineer with the Interborough Rapid Transit Co., New York City.

Walter C. Lange has resigned his position of engineer's assistant with the Consolidated Gas Co., of New York, and has accepted the position of superintendent of the Schmitt Manufacturing Co., Homestead, N. J.

F. W. Ballard, under whose direction the City of Cleveland installed the largest municipally owned electric station yet built, has been elected to the presidency of the Cleveland Engineering Society for the year commencing June, 1916.

Electrical Engineers of Six Cities Hold Joint Meeting

That time and distance no longer form a barrier to unity of action was fittingly illustrated last Tuesday evening by the national meeting of the American Institute of Electrical Engineers. To commemorate the achievements of engineering as a factor in modern civilization, the electrical engineers of Atlanta, Boston, Chicago, New York, San Francisco and Philadelphia, to the number of over 5,000, were brought together in one great meeting by means of long-distance telephone. A receiver was attached to each chair, and the speakers, scattered over the Union, were heard by all present.

The meeting was called to order by J. J. Carty, president of the Institute, who referred to the growth of electrical engineering and characterized the event as denoting the end of sectionalism. He read a telegram from President Wilson commending the engineers for their work in the development of the country's resources. The secretary, F. L. Hutchinson, then called the roll of cities, San Francisco responding with 650 present, Atlanta with 700, Philadelphia and Boston each with approximately 950, Chicago with 1,000 and New York with 1,100.

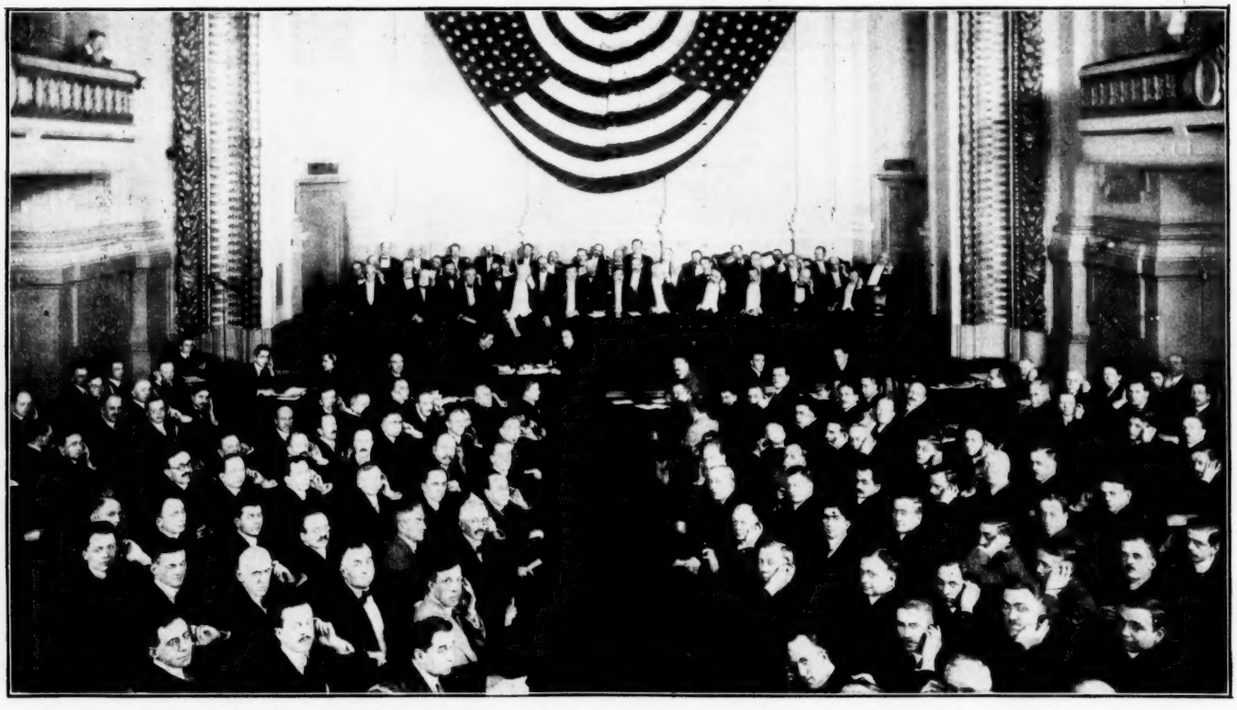
The report on the election of officers was read, showing those for the coming year to be: President, H. W. Buck; vice-presidents, L. T. Robinson, Peter Junkersfeld and B. A. Behrend; managers, J. B. Fisher, Charles Robbins, N. A. Carle and C. S. Ruffner; treasurer, G. A. Hamilton; secretary, F. L. Hutchinson. President-elect Buck then offered a few remarks in which he solicited the coöperation of the organization and governing body for the coming year, and said that President Carty had established a record which it would be difficult for his successor to approach.

Among the notables on the rostrum none received a greater ovation than Alexander Graham Bell, inventor of the telephone, who expressed great satisfaction at having lived to see the consummation of his invention; but he confessed that the telephone with its present complexities had now got beyond him. Greetings were also offered by Theodore N. Vail, who congratulated the Institute on its growth and spoke of the credit due the pioneers in the profession; by Thomas A. Watson, an associate of Doctor Bell; and by C. Le Maistre, who represented the British engineering societies.

Addresses were then delivered to the audiences of the different sections by noted educators, as follows, New York, Dr. John H. Finley, president of the University of the State of New York; Boston, Dr. A. L. Lowell, president of Harvard; Chicago, Dr. H. P. Judson, president of the University of Chicago; Philadelphia, Dr. E. F. Smith, provost of the University of Pennsylvania; Atlanta, Dean C. E. Ferris, of the engineering faculty of the University of Tennessee; and San Francisco, by Dr. R. L. Wilbur, president of Leland Stanford Jr. University. Doctor Finley referred to the pioneer work of Prof. Joseph Henry in electromagnetism, particularly the first electromagnetic motor and the first telegraph instrument, and exhibited some of these early experimental models. He dwelt upon what engineering had accomplished for the comfort and convenience of mankind and prophesied still greater inventions in the future.

Following greetings from the presiding officers of the sections participating, national airs from phonographs in the different cities came over the wires, to the tune of "Dixie" from Atlanta, "Yankee Doodle" from Boston, "America" from Chicago, "Hail, Columbia," from New York, "Columbia, the Gem of the Ocean," from San Francisco and the "Star Spangled Banner" from Philadelphia.

The meeting concluded with remarks on "The Engineering Profession," by Dr. Michael Pupin and Honorary Secretary Pope, the motion to adjourn being offered by Chicago, seconded by Atlanta and put to the entire meeting, the response coming simultaneously over all the wires.



NEW YORK SECTION OF NATIONAL MEETING OF AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Recent Legal Decisions

Appropriation of Water-Power Rights—In the absence of statutory regulations to the contrary, priority of water-power rights in a stream is accorded to that company which first defines and marks its route and adopts the same as its permanent location by authoritative corporate action. A water-power company constituting a public utility is not deprived of its right to condemn property for public use by reason of the fact that the company is also empowered to transact private business. (North Carolina Supreme Court, Carolina-Tennessee Power Co. vs. Hiawassee River Power Co., 88 "Southeastern Reporter," 349.)